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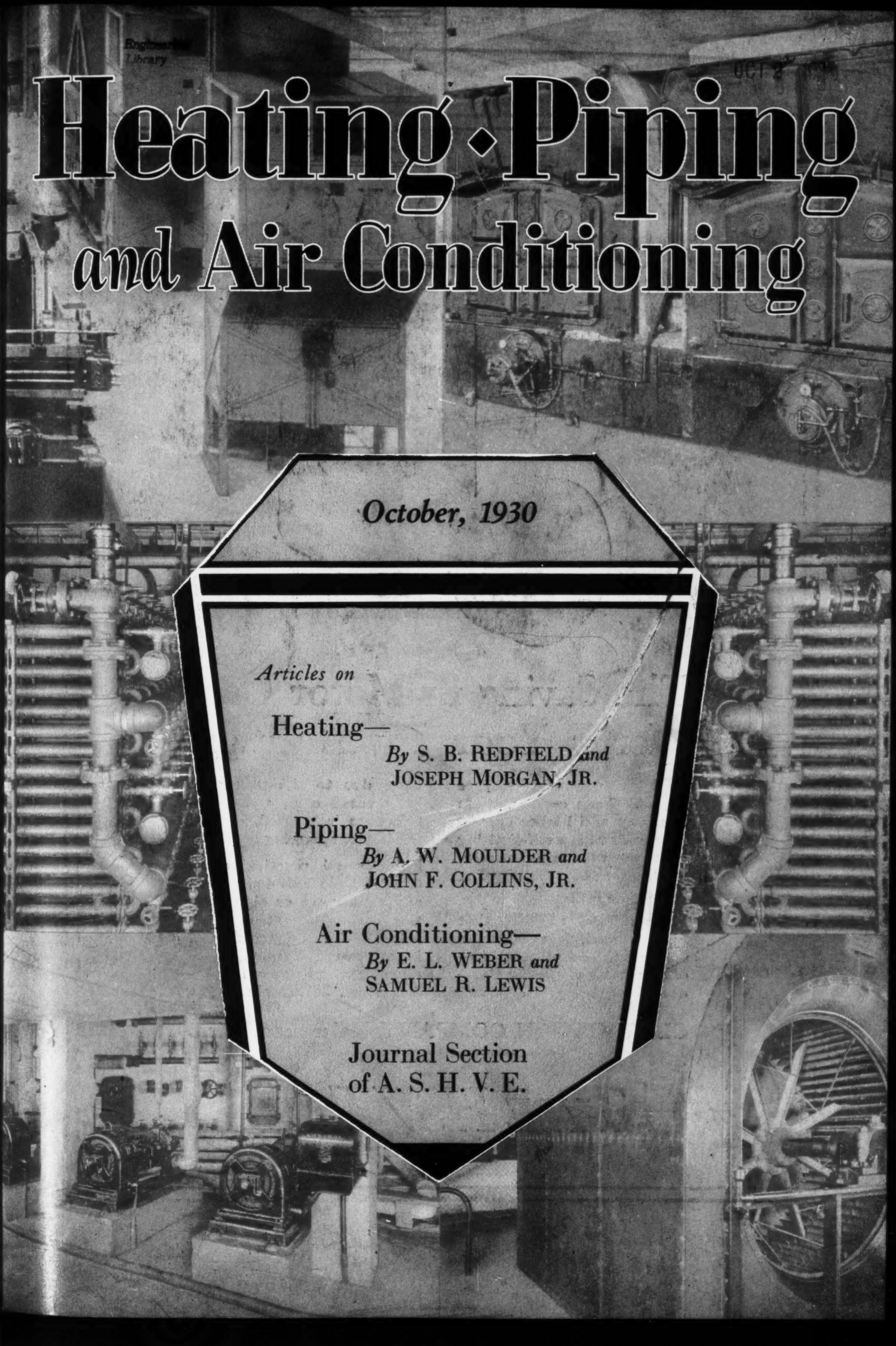
Heating · Piping and Air Conditioning



October, 1930

Articles on

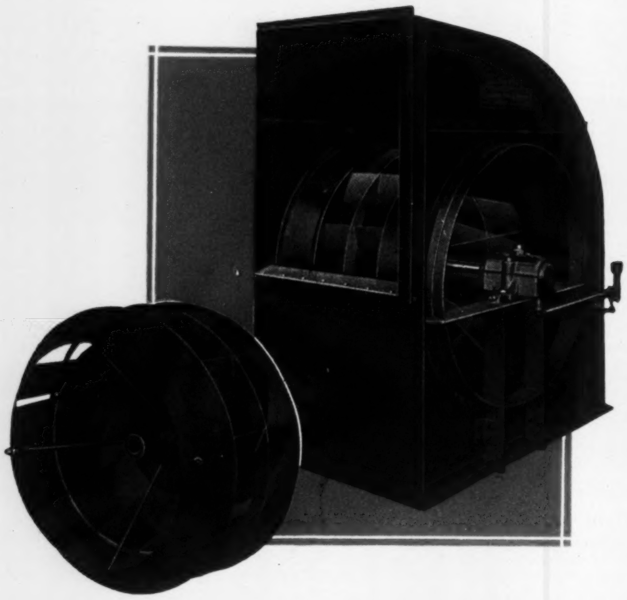
Heating—
By S. B. REDFIELD *and*
JOSEPH MORGAN, JR.

Piping—
By A. W. MOULDER *and*
JOHN F. COLLINS, JR.

Air Conditioning—
By E. L. WEBER *and*
SAMUEL R. LEWIS

Journal Section
of A. S. H. V. E.

CLARAGE



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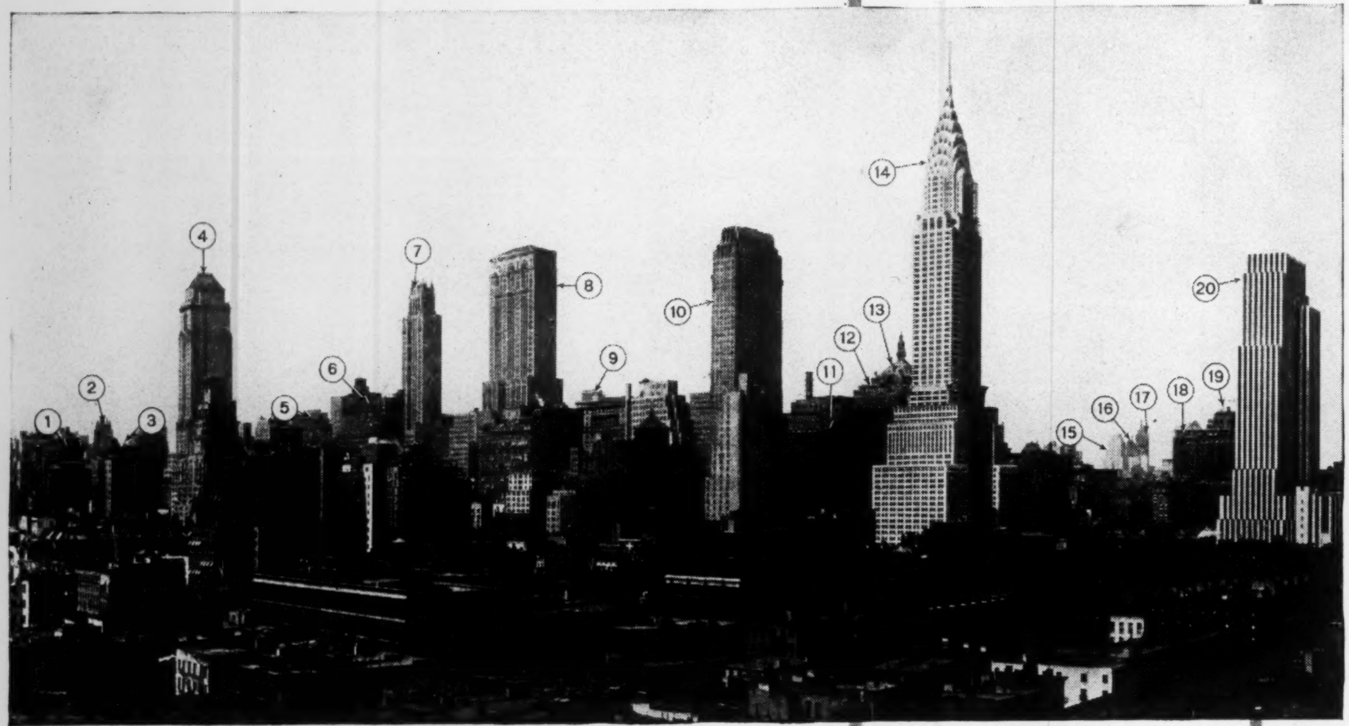

Panoramic view of mid-Manhattan, New York City's new skyscraper section.

in these sovereigns of the skyline

Architects, engineers and owners, in specifying heating pumps, sewage ejectors, house service pumps and sump pumps for fine buildings, turn naturally to Jennings. They know that Jennings equipment can be depended upon for years of efficient, trouble-free service.

In the majority of the towering structures visible in the above illustration, Jennings Pumps are installed.

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6. Murray Hill Building
   1 Duplex Jennings Vacuum Heating Pump
7. Lefcourt-Colonial Building
   1 Duplex Jennings Vacuum Heating Pump
8. Lincoln Building
   1 Jennings Dnplex Vacuum Heating Pump
   1 Jennings Vacuum Heating Pump
   2 Jennings Condensation Pumps
   1 Duplex Jennings Pneumatic Sewage Ejector
   1 Duplex Jennings Suction Sump Pump
9. Delmonico Building
   1 Duplex Jennings Vacuum Heating Pump
10. Chanin Building
    3 Jennings Vacuum Heating Pumps
11. Commodore Hotel
    2 Duplex Jennings Vacuum Heating Pumps
12. Graybar Building
    3 Jennings Vacuum Heating Pumps
13. N. Y. Central Building
    2 Duplex Jennings Vacuum Heating Pumps
    1 Jennings Condensation Pump and Receiver
14. Chrysler Building
    2 Duplex Jennings Vacuum Heating Pumps
    3 Jennings Suction Sump Pumps
    1 Duplex Jennings Pneumatic Sewage Ejector
    1 Duplex Jennings Condensation Pump
15. Squibb Building
    1 Duplex Jennings Pneumatic Sewage Ejector
    1 Duplex Jennings Vacuum Heating Pump
16. Savoy Plaza Hotel
    4 Duplex Jennings Vacuum Heating Pumps
    2 Jennings Suction Sump Pumps
17. Sherry-Netherland Hotel
    4 Jennings Pneumatic Sewage Ejectors
    2 Jennings Condensation Pumps and Receivers
    1 Duplex Jennings Vacuum Heating Pump
18. Bartholomew Building
    1 Duplex Jennings Vacuum Heating Pump
19. Shelton Hotel
    2 Duplex Jennings Vacuum Heating Pumps
    1 Jennings Vacuum Heating Pump
20. Daily News Building
    2 Duplex Jennings Vacuum Heating Pumps
    1 Jennings Vacuum Heating Pump

Heating · Piping and Air Conditioning

Vol. II October, 1930 No. 10

Small Variations Allowable *in* Air Conditioning *for* Gelatin Capsule Manufacture

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Eight Sizes of Capsules Made in the Eli Lilly Plant

By William A. Hanley*

Until air conditioning became a practical science, gelatin capsules were never manufactured in any plant during the summer months, and with only limited success during winter months. The present day large-scale operations are entirely dependent on proper air conditioning. The best information the writer can obtain leads to the conclusion, that the first scientifically controlled modern air conditioning unit was built for drying photographic film and the second unit for drying gelatin capsules.

There have never been more than five or six gelatin capsule plants operating in the world. This is due largely to the highly specialized character of the product and to the intricacies of the process of manufacture. Capsules are molded on bronze pins in automatic capsule manufacturing machines. The function of the machine is to take a bar, containing twenty to thirty bronze pins, dip this bar in molten gelatin, withdraw the bar so as to obtain a uniform gelatin film, revolve it so as to avoid flowing of the gelatin, dry the gelatin film and then feed the bar to a second part of the machine. In this second part of the machine, all automatic, the capsules are removed from the molding pins, trimmed to uniform length, joined together and dropped from the machine, the capsule bars continuing on to repeat the cycle.

Thickness of Capsule Wall

When it is remembered that the thickness of the capsule wall is only three and one-half to five thousandths of an inch, that the variation of wall thickness for each particular size of capsule can not exceed one thousandth of an inch, that capsules are highly sensitive to moisture changes, and that they have a rather fragile body to be handled by machinery, it is realized that their manufacture probably represents the most exact molding carried on in the world today.

The air conditioning plant for gelatin capsules is, of course, designed primarily for the purpose of drying the capsules after they are dipped on the pins. The plant, however, performs a number of other functions, all of which are necessary for successful manufacturing. One of these functions is that the pins before they are dipped in the gelatin must be at a fixed uniform temperature. Should the pins be too warm, the gelatin will flow by gravity down the capsule pins, giving a capsule wall thick at one place and thin at another. Should the pins be too cold when dipped, the capsule walls will be uniformly too thick, and the capsule can not be joined, as the cap or short part of the capsule will not fit over the body or long part of the capsule. This temperature range of the pins on which capsules are molded can not exceed one to two degrees Fahrenheit. The air conditioning unit maintains the pin temperature, which results in uniform thickness of capsule walls. Of course, the gelatin solution must have a constant temperature and fixed viscosity, but these are apart from the air conditioning.

Eight Different Sizes

Capsules for human use are made in eight different sizes, the smallest size having a capacity of approximately one-sixteenth the volume of the largest size.

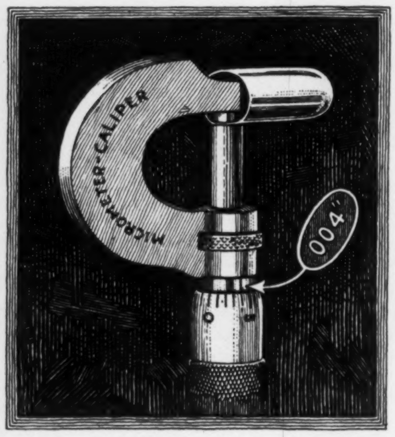
---

* Eli Lilly & Company, Indianapolis, Ind.

The larger capsules require a longer drying period with air having a lower relative humidity. To take care of this condition, the air conditioning plant supplies all the air at the lowest temperature and humidity required for any size capsule. The machines are arranged in batteries according to size on the main supply duct with a branch duct to each battery. In each branch duct, the air is further heated by thermostatically controlled steam coils for supplying just the temperature required for each size of capsule. Likewise, vaporizers supply any extra moisture needed.

Moisture Content Can Not Vary

The wall of gelatin of a capsule is so delicate that the limits of air quantity, temperature and humidity are all very close. Particularly for the smaller sizes of capsules the air quantity can not vary more than five per cent, while the temperature should not vary more than one degree Fahrenheit from a fixed point. The moisture content of the air must not vary over one-fourth of a grain per cubic foot if a satisfactory product is to be continuously produced.



Capsule Wall Thickness Is Only .0035 to .005 In.

The Air Conditioning Plant

The air conditioning plant for manufacturing capsules is a humidifying plant in winter and a dehumidifying plant in summer. It, of course, is true that the outside air conditions are just right at certain times, and that the air conditioning plant has no work to do. However, such days number about one dozen in a whole year and usually only two or three hours on such days. For continuous operation of the capsule factory, the air conditioning equipment must be operated daily. The hours it will be operated when not needed will pay big insurance on the decreased output caused by a shut down due to unsatisfactory air conditions.

The air conditioning plant for manufacturing capsules must be operated in winter about one hour and from two to four hours in summer before the dipping of capsules is started and usually must run one-half hour after the last capsules are dipped. The morning operation, of course, is to produce a uniform temperature in the room, in the drying kilns, and in the bronze molding pins. The operation after dipping is to complete the drying of capsules dipped near the end of the day.

Construction of Plant

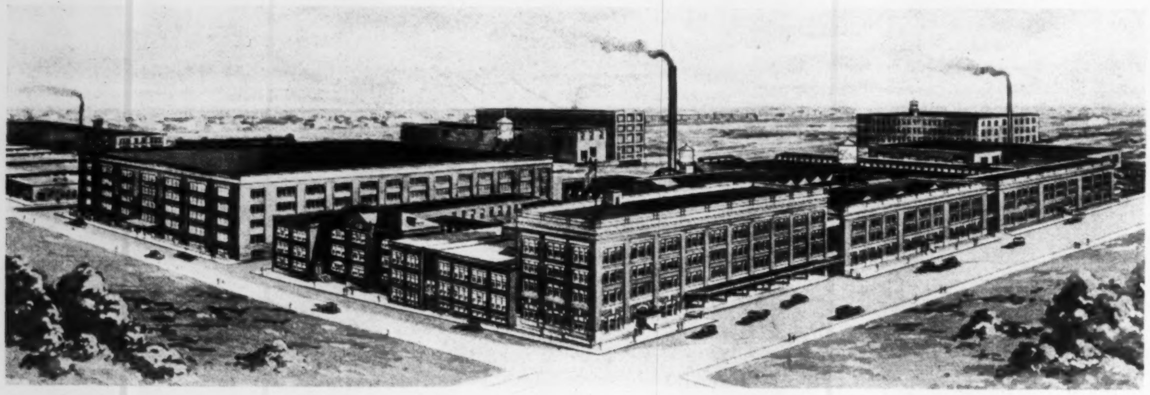
The room in which capsules are manufactured is usually built with hollow tile plastered walls, double windows, weather-stripped, vestibules at each entrance door, and with double floors having heavy hardware paper or roofing paper between floors. These precautions are taken to stop air infiltration and to reduce the load on the air conditioning plant in humid weather.

The air conditioning plant is usually located in a room below or in a building adjoining the capsule manufacturing plant. When the air conditioning plant must be located some distance from the capsule manufacturing plant it has been found that concrete ducts below ground, properly protected against water, make a very satisfactory installation.

Washer Insulated

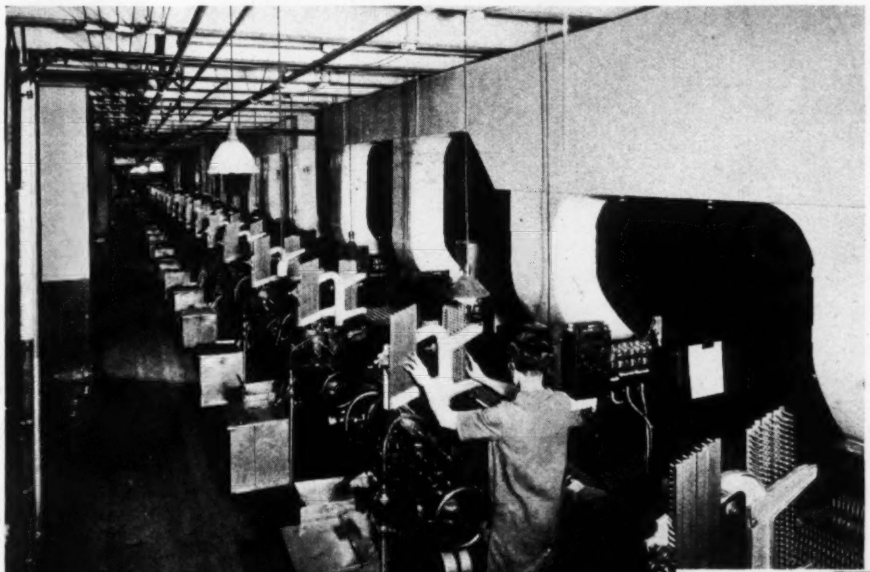
The air conditioning equipment is of the conventional type. Because of the fact that saturation of the air must occur at temperatures of 40 to 50 below summer weather conditions, two and sometimes three rows of spray nozzles are used in the washer. Since the circulating water for the washer will be somewhat colder than the air leaving the washer, it has been found economical to insulate the washer with cork board or some other such covering. As the air exhausted from the drying kilns is both colder and dryer than ordinary outside summer air, the air conditioning plants are of the recirculating type.

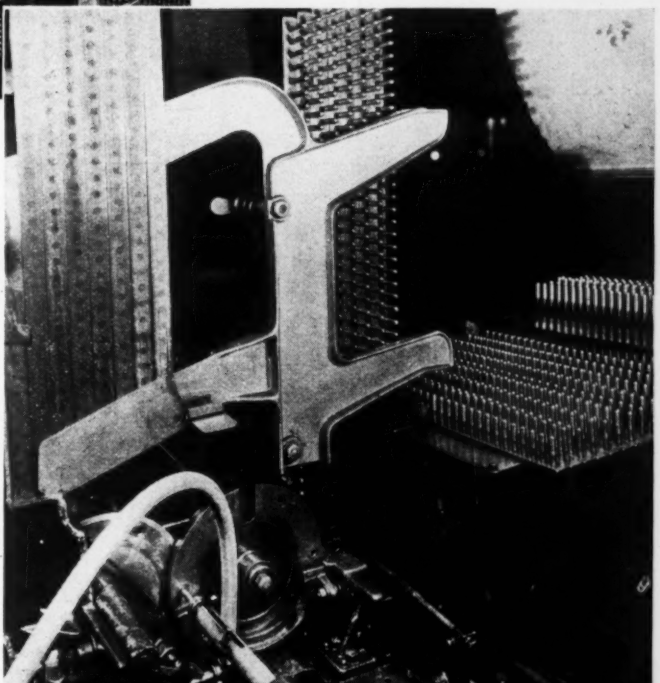
One of the economies ordinarily taken advantage of is the use of an interchanger. This is made possible because the air can not be used as cold as it leaves the washer. The interchanger heats the air leaving the washer by reducing the temperature of the air being returned from the manufacturing room, which is to be recirculated through the washer. Sufficient fresh air must be introduced to keep the conditions in the manufacturing plant free from an excess of carbon dioxide, but this is hardly a problem, since infiltration, through the opening of doors and in many other unseen ways, supplies sometimes much more than the required fresh air.



The Indianapolis Plant of Eli Lilly & Co. Other Plants and Branches Are Located at Greenfield, Ind., New York, Chicago, New Orleans, Kansas City, St. Louis, and Shanghai, China

Due to the fact that dehumidifying requires refrigeration, and that dehumidifying is not carried on in the winter months, a good heat balance can be made by using exhaust steam refrigerating machines for cooling the water being circulated through the washer. The exhaust steam can be used for heating buildings in the winter and for operating refrigerating machines in the summer after it has already converted some of its heat into mechanical energy; driving turbines, engines, air compressors or some other such equipment.





LEFT—A ROW OF AUTOMATIC CAPSULE MACHINES. NOTE THE LOG CHART ON THE SIDE OF THE KILN WHERE THIRTY-MINUTE READINGS ARE RECORDED. BELOW—A CLOSE-UP OF THE STRIPPING, JOINING AND CUTTING MECHANISM

Filter at Discharge of Recirculating Fan

In some instances the entire quantity of air used in drying capsules is returned and put through the dehumidifier; while in other installations only about one-half the air is dehumidified, the balance being simply recirculated either by small fans in the manufacturing room or by large fans in connection with the air conditioning equipment. Gelatin in a molten state is very sticky, so that dust particles can do immense damage. The air passing through the washer of course comes out very clean, but air recirculated may pick up much dust in the manufacturing room from operators' clothes, shoes and from the floor. A satisfactory method of taking this from the recirculated air is with an air filter of the continuous operating oil type. The scheme has been found effective and the filter is ordinarily located at the discharge of the recirculating fan, as from that point to the machines the system is under pressure, so that leakage will be outward and thus no dust will be sucked in. While all the above is true, every plant handling gelatin and using air must be thoroughly cleaned up three or four times per year. Small dirt particles will creep in and only a thorough wiping of all surfaces of the duct system will prevent capsules from becoming dust-specked.

Comfortable Air Conditions Reduce Labor Turnover

The employes in a capsule plant are very fortunate in that the summer working conditions are highly invigorating and the humid hot days do not discomfort the workers. In winter, of course, the air in the capsule plant contains much more moisture than will be found outside, so that there is no drying and cracking of the skin. The machine operators being girls, these air conditions being so satisfactory, means that capsule plant jobs are much sought after and labor turnover is at a minimum.

The modern air conditioning plant for capsule manufacturing had its beginning about twenty to twenty-five years ago. It took two or three years to work out all the intricacies of the application. During these two or three years, there was a steady increase in output per employe, a better and more uniform product, a predetermined fixed output for each working day, and more capsules produced for each pound of gelatin used. Each of the above factors contributed to making the monetary returns on the air conditioning equipment more attractive. With a demonstration of what had been accomplished on capsules, the drug and chemical manufacturers were quick to apply air conditioning to the manufacture of tablets, pills, chemicals and all products which were hygroscopic in nature or required changes of moisture content.

Fern Rock Car Shops Serve Philadelphia Subway—Mechanical Equipment of Interest

By Joseph Morgan, Jr.*

AN IMPORTANT part of Philadelphia's new subway system is the Fern Rock yard and repair shops, several squares beyond the northern passenger terminal, where all of the cars are periodically inspected and overhauled to insure safety and comfort in transportation.

With an eye to future transportation problems, when Broad Street throughout its whole length will be included in Philadelphia's subways, the car shops were most carefully planned by the department of city transit to permit proper maintenance of the large number of cars which will be in use at that time. Part of this planning included the design of the heating systems to satisfactorily warm the two main divisions—the inspection shop and the general repair shop.

Inspection Shop

Cars routed to the yard for other purposes than storage first enter the inspection shop where workmen, stationed below and between the rails in the inspection pits, carefully examine them for defects or signs of wear. There are five pits, each of which will accommodate a train of six cars.

In order that the men may work in comfort during winter weather, with the large steel bodies and parts (which are cold) directly above them, warmed air is furnished by a hot blast system and discharged directly into the pits through registers in the side walls, placed at 20-ft. intervals.

Between the pits are platforms which conveniently serve as plenum chambers. Fans and heaters are located in the basement of the adjacent general repair shop and consist of two 13,000 c.f.m. fans delivering through individual underground ducts to the plenum chambers. A common return duct, connected to the space below the middle platform at the inspection shop end and to the fan room in the repair shop, serves for recirculation. Two stacks of heaters are required for each fan, and warm the air from 40 to 100 F.

Below all windows on the long side walls are double tiers of wall radiators, each tier separately valved, and supplied from an overhead steam main by drop risers, both pipe and radiator in each bay being fitted well within the projection of the pilasters. Hand control of the radiators is used throughout, and both (or either) hot blast fans may be used, as required for variable weather conditions.

Repair Shops

Under the heading general repair shops comes a group of departments under one roof which is used entirely for maintenance. Following inspection, the cars are scheduled for repairs or replacements and are given a thorough overhauling.

In the west section is the truck shop, with bearing, wheel, machine, welding, and forge shops adjoining on the first floor, and overhead on the open mezzanine floor is a long section where bench work is done.

Across the transfer table, on the east side, are departments for jacking-up, electrical repairs, air brakes, overhauling, washing, cabinet and upholstery, and painting. Additional space is provided for offices, instruction, lockers, first aid, cafeteria, wash-rooms and storage. In the last named group, direct radiation alone is used.

THE SHOPS FROM THE REAR. AT THE LEFT IS THE INSPECTION SHOP; THE POWER PLANT WITH THE ASH CONVEYOR AND RADIAL BRICK CHIMNEY IS SHOWN IN THE MIDDLE OF THE PICTURE. IN THE FUTURE, REPAIR SHOPS WILL BE ENLARGED GIVING 50,000 SQ. FT. ADDITIONAL FLOOR SPACE

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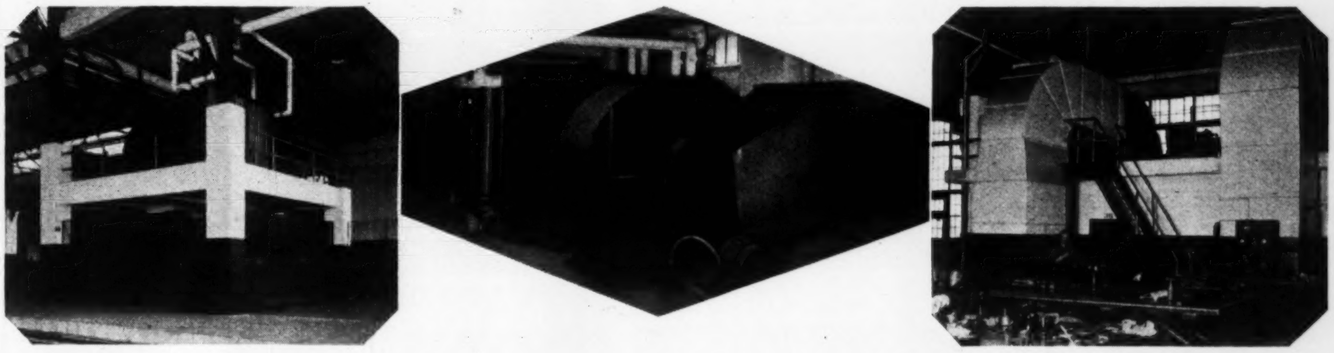
* Stewart A. Jellett Co., Philadelphia.

The remainder of the building is heated with both radiators and hot blast except in the paint shop, where coils along the middle row of columns and wall radiators along the side walls (no exposure), all hung eight feet from the floor, are used to keep a uniform temperature.



Left—The Third Fan and Heater Unit Is Mounted on a Platform at the Northwest Corner of the General Repair Shop. The Air Discharge Duct, with Canvas Joint, Connects Into a Concrete Distribution Duct Beneath the Floor Below. This View Was Taken from the North End of the Transfer Table Looking Southwest Toward the Fan

Center—Fans and Heaters for the Inspection Shop Are Located in the Basement of the Adjacent General Repair Shop. The Arrangement Is Symmetrical, Using Left and Right Hand Fans

Right—These Are the Largest Fans in the Shops and an Idea of Their Size Can Be Obtained by Comparison with the Figure of the Man at Work on the Floor Below. They Are Located Near the East Wall of the General Repair Shop

This department can be entirely closed off from the others by folding doors at the transfer table end where the cars enter. As the car washing and upholstery sections have no floor pits, warm air from the heaters reaches them only by "accident."

The hot blast system in the general repair shop differs from that in the inspection shop in its method of return circulation. Air from the four sets of fans and heaters is discharged through registers in the side walls of the floor pits but it returns unrestricted by any collecting ducts, across the entire room space, above the floor to the open side of the heater casings.

The heaters and fans are located on platforms ten feet high and so the warm air leaving the pits takes a circuitous path back to the fan intakes, thoroughly permeating the building and maintaining uniform heat throughout.

The east group receives 68,000 cubic feet of air per minute from two fans which discharge down through sheet metal ducts into concrete interconnecting ducts below the floor leading to the plenum chambers.

The west group is heated by 68,000 c.f.m. from two fans which deliver in a similar manner, although here the fans are placed at opposite ends instead of both together near the middle.

The Boiler Plant

The boiler plant is located in the rear of the repair shop. It was placed there for convenience, but, unlike many another plant "stuck in the back yard," it operates most efficiently and conveniently.

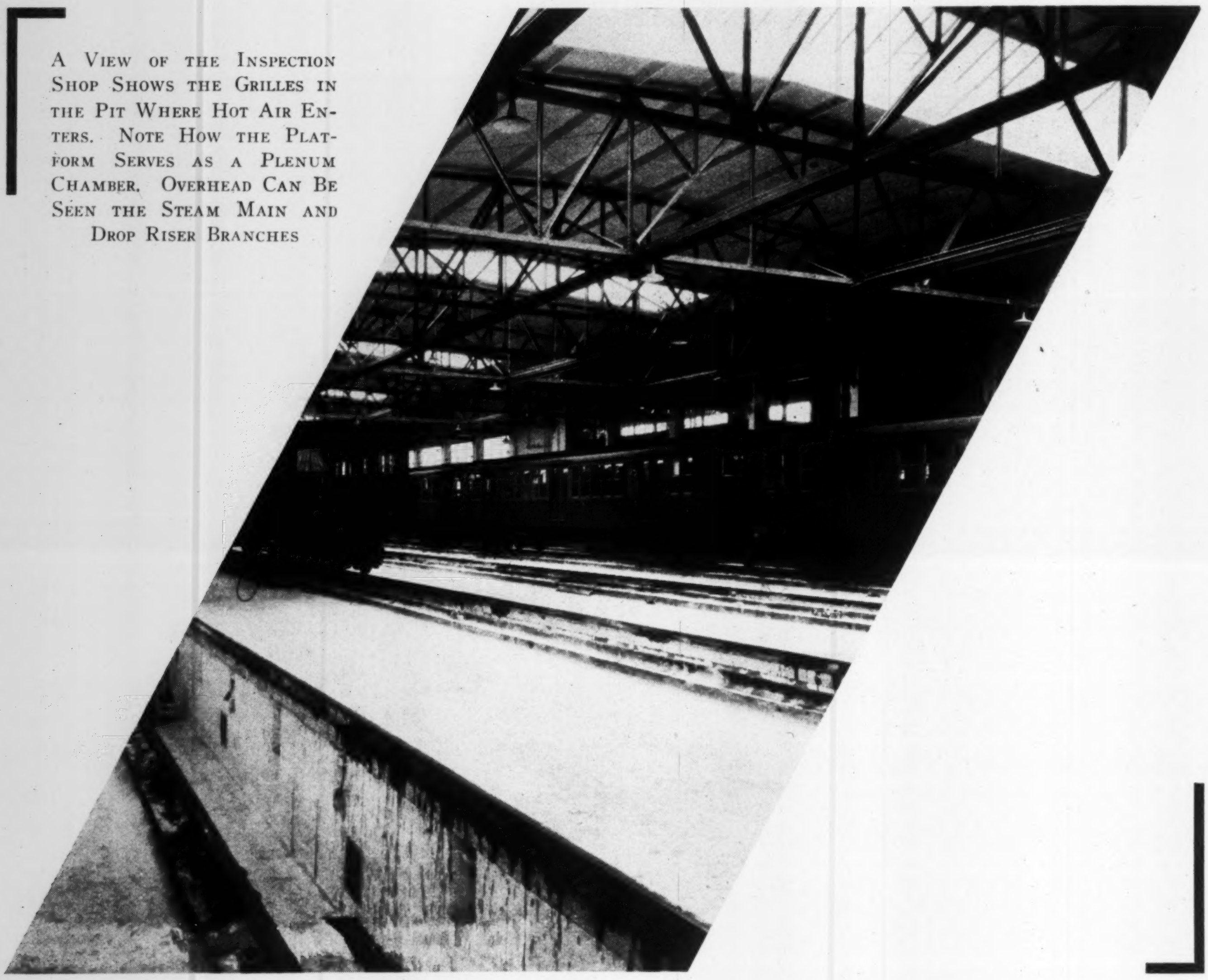
Two 333 hp. long-drum, water-tube boilers, in battery setting, handle the present 560 hp. load with considerable ease and flexibility in operation, and have excess capacity for future extension of the buildings.

They were built for 160 pounds but operate at 15 or less. Space limitations made the tube dimension choice 16 wide by 10 high. At present they are hand-fired with bituminous coal but the headers are set high enough for future stoker installation, and provision is made for two 8,500 c.f.m. forced draft fans to connect to the ducts now built into the ash pits. Later, when the increasing traffic makes necessary the extension of the inspection and general repair shops, 200 hp. will be added to the heating load.

To provide steam for heating the offices, cafeteria, lockers, toilets and entrance building during spring and fall seasons of the year, and to supply hot water during the summer months, a small auxiliary boiler is installed. It has 5,700 square feet of surface, of which 3,000 is required for hot water heating. The schedule calls for it to be fired on April 15th, at which time the large boilers are shut down. Both plants are operated at low pressure of about 2 to 5 pounds.

Plenty of draft is provided for both the steel boilers and an incinerator by a six-foot diameter radial brick

In the Paint Shop Section of the General Repair Shop Wall Radiators and Coils Were Kept 8 Feet Above the Floor. This Part Can Be Shut Off From the Other Section of the Repair Shop by Folding Doors



A View of the Inspection Shop Shows the Grilles in the Pit Where Hot Air Enters. Note How the Platform Serves as a Plenum Chamber. Overhead Can Be Seen the Steam Main and Drop Riser Branches

stack which starts the first course of its 150 foot height eleven feet above the boiler room floor.

Ash Removal

Ashes are shovelled into a grating at the boiler room floor level near the boilers, where a 7-ton hoist of the vertical centrifugal discharge bucket type carries them from the hopper, 54 feet up to a storage bin above the roof. Chutes from this bin permit the same car that delivered coal to be used for hauling away the ashes, or a separate pair of chutes allows a truck to pass directly below the bin to receive the ashes.

Adjacent to the boiler room is the pump room containing two motor-driven vacuum pumps and a 900 gallon-per-hour hot water generator with its circulating pump below.

The larger vacuum pump is a 40 pound 100,000 sq. ft. duplex unit connected to the return piping from heaters, radiators and the hot water generator. An auxiliary pump is used in conjunction with the small boiler.

Boiler Headers

Above the large boilers is a 14-inch header from which extend two 10-inch mains to the shops and a 5-inch cross to the header of the auxiliary boiler.

From the auxiliary boiler header a 4-inch main travels along through the shops close to the larger pipe and supplies those radiators which are used for spring and fall heating. A separate vacuum main returns the condensate from such radiators to the small pump and discharges to the cast iron boiler.

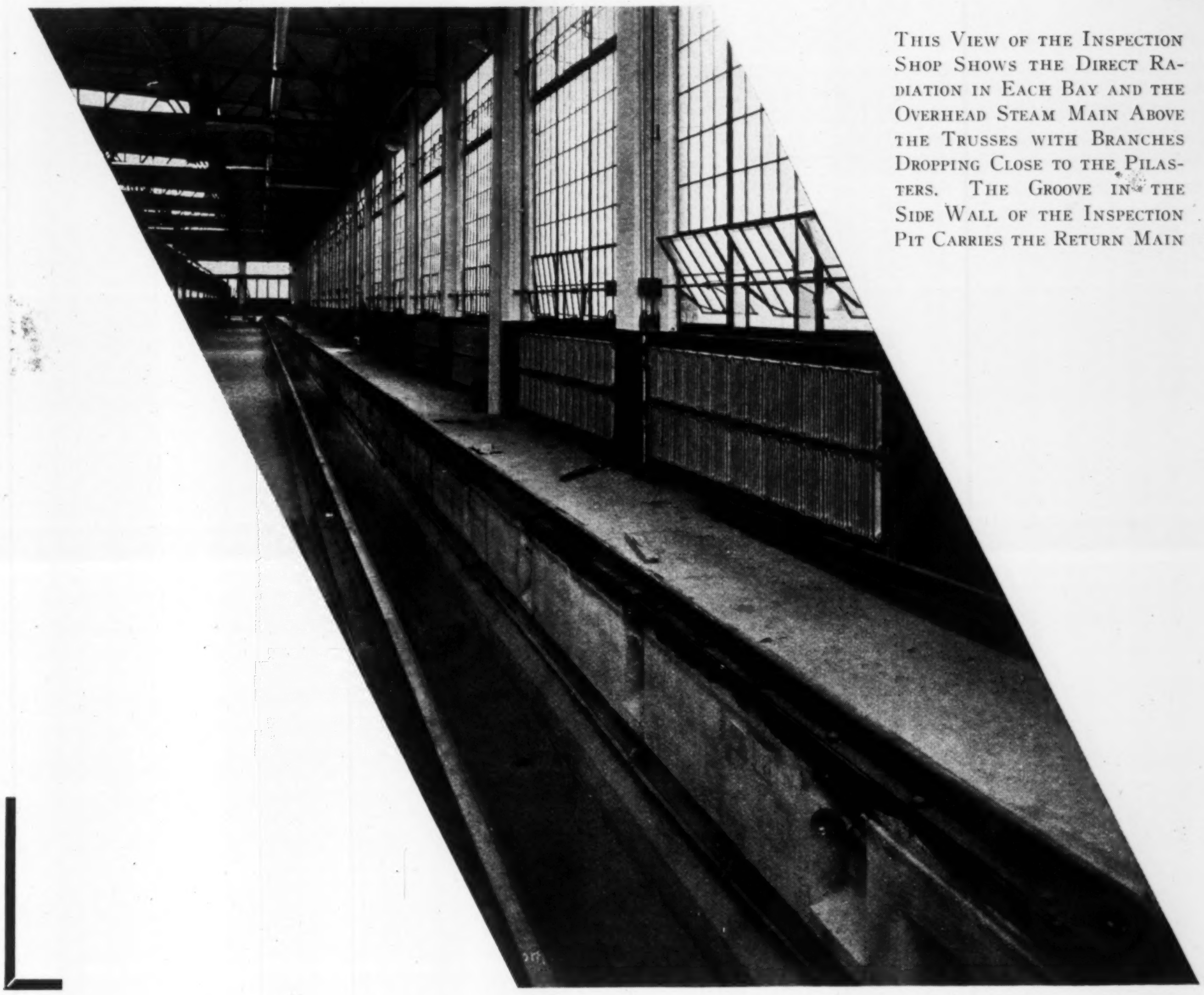
The lines to the entrance building extend underground in a tile conduit to the repair shop 250 feet distant. Double-slip expansion joints in the lines are placed in a manhole midway between the buildings and are the only steam pipe expansion joints used throughout the plant. Expansion bends were used where required, approximately at 100 foot or less intervals, and the mains were anchored in the middle of the bends. Where return mains required expansion joints, manholes were provided for easy access to the joints.

The most difficult problem with the overhead steam lines was to locate them clear of crane travel. In general they were carried above the lower chord of the roof trusses or against the walls back of the tops of the columns.

All pipes except exposed returns were covered with 85 per cent magnesia sections and extra sewed canvas jackets. In addition, the underground return mains were encased in terra cotta. In the inspection pits, returns were fitted into grooves in the side walls.

Design of Incinerator Smoke Flue

To take care of the rubbish which accumulates at the terminal point where cars are inspected and cleaned, an



This View of the Inspection Shop Shows the Direct Radiation in Each Bay and the Overhead Steam Main Above the Trusses with Branches Dropping Close to the Pilasters. The Groove in the Side Wall of the Inspection Pit Carries the Return Main

incinerator with a capacity of 1,000 pounds of light refuse per hour was installed. Material for incineration can be easily wheeled on the charging platform at ground level and dumped into two chutes. The small quantity of ashes obtained are placed in the conveyor in the boiler room nearby.

Draft Loss in Stack Corrected

The incinerator has a bypass flue for overload and a main flue, both equipped with dampers. The main flue damper at first did not fit tightly and caused considerable draft loss in the radial brick stack into which the incinerator flue vents. This trouble was discovered during the boiler test and was remedied by decreasing the clearance of the damper in the slot. The leakage was severe only when the incinerator was not in operation, probably because during combustion of refuse, the hot gases compensated by increasing the draft.

As the incinerator is at one side of the boiler room at a considerable distance from the stack and with an intervening passageway, the design of the smoke flue was a real problem. The drawing on page 816 shows how it is built below the floor to clear the passage, rises 17 feet to ground level, and heads toward the stack, where, in the center it forms a vertical neck to keep the flame from lapping the inside walls of the chimney. This flue is 70 feet long, entirely firebrick lined, and well supplied with clean-outs.

Calculating Fan Sizes

To determine proper fan sizes the procedure was as follows: Total Btu loss per hour of the building was calculated for maintenance of a predetermined inside temperature. From this total was deducted the equivalent radiating rate of the direct surface to be installed, leaving the net amount of heat to be supplied by the warm air. Fixing 100 degrees as the highest temperature at which it was desired to have the air enter the pits, and keeping the friction through the heaters within reasonable limits determined the quantity of air, recirculated at 40 degrees, required to supply this net heat. To permit flexibility in operation, the air supply was divided among the proper number of fans, which in the case of the inspection shop was two. Duct and grille sizes were based on velocities of 300 and 800 feet per second, respectively. The air velocity through the pit openings was kept low so that the workmen close to it would not be made uncomfortable.

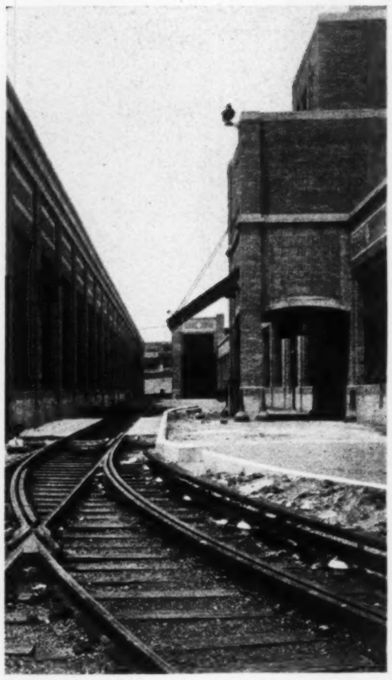
Heating calculations were made by the *Heating and Piping Contractors' National Association* method, and checked by a short-cut rule based on a refiltration method. The base temperature for Philadelphia is 15 degrees, and calculations were made accordingly.

The inspection shop was figured for heating to 55 degrees, other shops for 60, offices, halls, cafeteria, lockers, wash rooms and toilets for 70, and paint and oil rooms for 80.

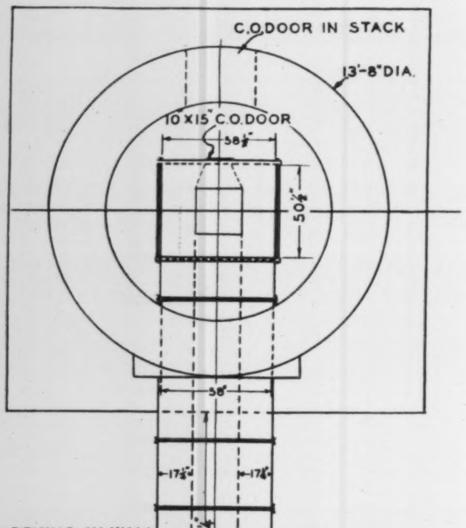
Of the total 560 hp. load, 320 was required for the direct radiation and the balance for the hot blast system which provides 130,000 cubic feet of air per minute.

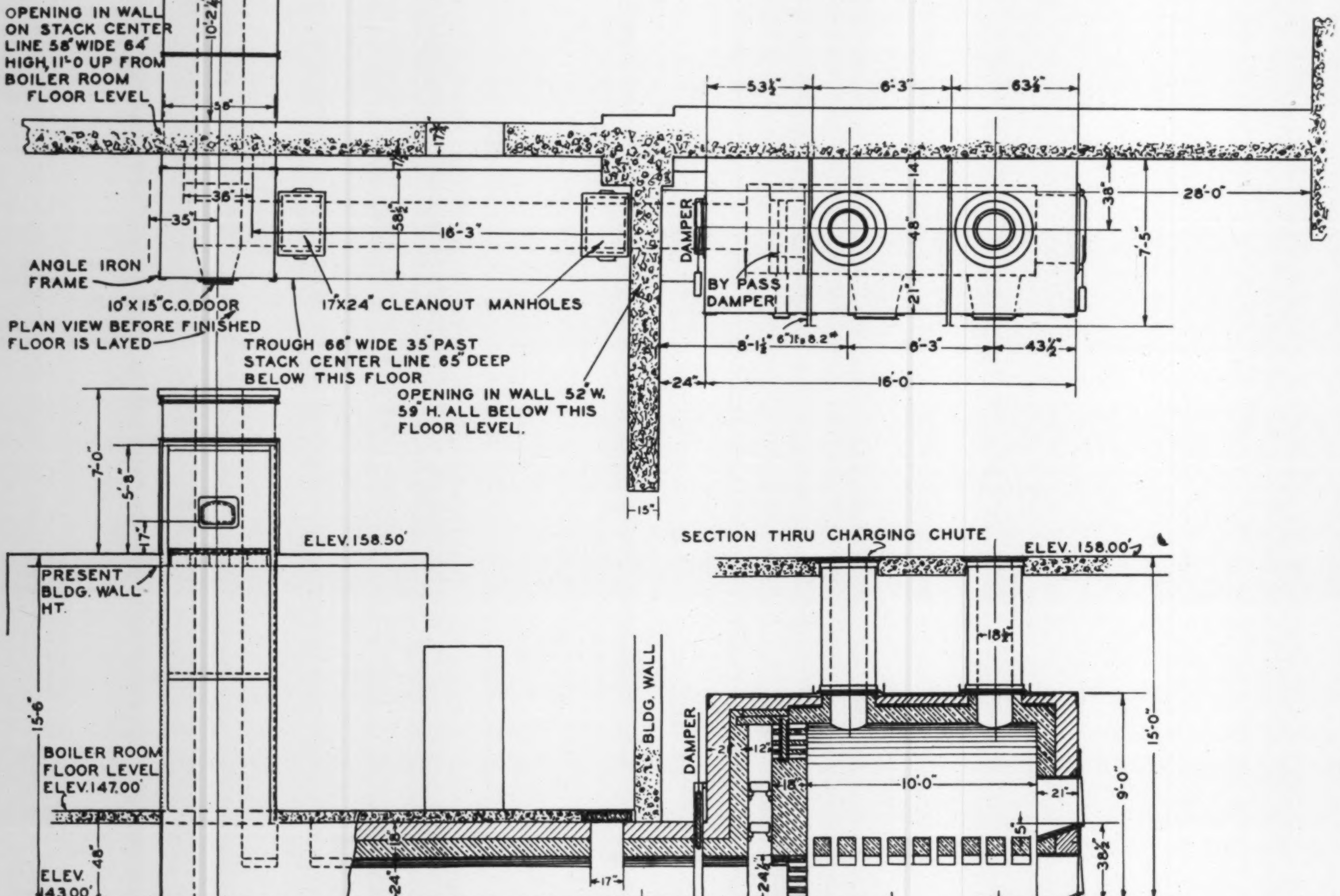
This project was probably most interesting because of the large area covered by the buildings and required to be heated. The heating system was designed by S. A. Jellett Company, and installed under joint supervision with the department of city transit.

At the completion of the work, the boilers were tested for capacity, and it is interesting to note that an overload of 50 per cent was obtained with hand-firing using No. 2 buckwheat coal. The entire system was then tested for capacity and circulation, and functioned perfectly in every respect.



THIS VIEW, TAKEN FROM THE REAR OF THE YARD, SHOWS THE GRATING WHERE COAL IS DUMPED FROM CARS INTO THE COAL POCKET BELOW. DIRECTLY ABOVE IS THE ASH CHUTE WHICH WILL DISCHARGE INTO CARS ON THE TRACK FROM THE CONVEYOR. THE UPPER PART OF THE BOILER ROOM IS SHOWN AT THE RIGHT. IN CASE TRUCKS ARE TO BE USED FOR ASH HAULING, THEY CAN DRIVE BETWEEN THE COLUMNS BELOW THE BIN WHERE A SEPARATE PAIR OF CHUTES ARE USED FOR EACH DISCHARGE


C.O.DOOR IN STACK
13'-8" DIA.
10X15" C.O.DOOR
58½"
50½"
58"
17½"
10'-2¼"


OPENING IN WALL ON STACK CENTER LINE 58" WIDE 64" HIGH, 11'-0 UP FROM BOILER ROOM FLOOR LEVEL
58"
17¼"
36"
35"
58½"
16'-3"
ANGLE IRON FRAME
10"X15"C.O.DOOR
PLAN VIEW BEFORE FINISHED FLOOR IS LAYED
17X24" CLEANOUT MANHOLES
TROUGH 66" WIDE 35" PAST STACK CENTER LINE 65" DEEP BELOW THIS FLOOR
OPENING IN WALL 52" W. 59" H. ALL BELOW THIS FLOOR LEVEL.
DAMPER
BY PASS DAMPER
53½"
6'-3"
63½"
14"
48"
21"
38"
28'-0"
7'-5"
8'-1½"
6" I's 8.2#
6'-3"
43½"
24"
16'-0"
15"
7'-0"
5'-8"
17"
ELEV. 158.50'
PRESENT BLDG. WALL HT.
15'-6"
BOILER ROOM FLOOR LEVEL ELEV. 147.00'
48"
ELEV. 143.00'
6"
70"
18"
24"
17"
9"
8"
23'-5
BLDG. WALL
DAMPER
SECTION THRU CHARGING CHUTE
ELEV. 158.00'
18½"
2"
12"
18"
10'-0"
21"
5"
38½"
9'-0"
15'-0"
24½"
18"
4½"
6"
17'-0"

THE INCINERATOR HAS A CAPACITY OF 1,000 POUNDS OF LIGHT REFUSE PER HOUR. THE ASHES ARE PLACED IN THE CONVEYOR IN THE ADJACENT BOILER ROOM

Determining the Location of District Heating Mains

John F. Collins, Jr.*

IN all the very large cities of the northern part of our country, in many of the medium-size ones and even in a considerable number of smaller ones there are today district heating companies serving steam from central generating stations to downtown buildings, and in some cases to outlying districts. Some of these companies are large, some are small, others are mature, and still others are just beginning to develop. In some of these, the engineers who decide where the lines are to be laid in the streets have been "through the mill" many times and need no further information on the subject to help them in their work. The writer hopes that the following recital of the problems arising in locating steam mains in his city, and the methods of solution, will be of material benefit to the young engineer and will at least be interesting reading to the older members of the profession.

The problem of making certain of selecting the best location beneath the streets of a city for pipe lines to transmit and distribute central station steam and to collect condensate return water, and the conduit or structure in which these lines are encased, is one which should be very carefully investigated. The study which enables the engineer to find the proper place for the lines is a paying one.

An unexpected obstruction, found in the path of a line during construction, may necessitate the relocation of one or the other at great expense. Suppose, for example, that a line is being built at such depth that the tunneling method of excavation has been found best. A shaft has been dug and a tunnel driven ahead for a hundred feet. A large sewer blocks the way. This cannot be raised or lowered. Therefore, it is necessary to deepen the excavated tunnel so the steam pipe conduit it is to contain will pass under the sewer. Extra expense results from the increase in quantity of earth to be removed and from the difficulty encountered in retimbering. A location which may on the surface seem the least costly and most desirable for line construction may prove under more minute examination to be impractical.

McCANN TUNNEL

Leaving out of the question the consideration of obstructions, a certain side of the street will usually be found best for construction, due to the size of present or prospective buildings on that side. Suppose a main is extended along the north side of a street, where only small structures can be built. On the south side large buildings are constructed, necessitating the extension of big service lines across the street. This may be avoided if the main is laid along the south side, as only small services or fewer large ones would be necessary for the small buildings on the opposite side of the street.

The amount of time required for construction is an important consideration. Service in a building is usually required by a certain date. The overhead and labor cost increase per unit of production, where delays occur. In Pittsburgh construction permits are usually given for work to be done only at night, because of the heavy traffic on the streets and sidewalks in the daytime. In order to inconvenience the public as little as possible, street openings have to be timbered over each morning and opened up each night. An increase on the number of times this is done adds to the cost. The permits specify a time limit within which the work must be done, and provide a fine for failure to complete it on schedule.

For these reasons a very complete survey is made before the construction of steam lines beneath the streets is attempted.

The following paragraphs explain the problems, the rules and regulations that must be followed and how these studies are gone about. However, the writer does not wish to leave the impression that each step mentioned is carried out on every job, for on each extension the conditions are not similar, and only such preliminary work is performed for each line as is considered necessary.

Municipal Regulations

The franchise area of the steam company comprises the downtown district, sometimes called the

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* Designing Engineer, Allegheny County Steam Heating Co.

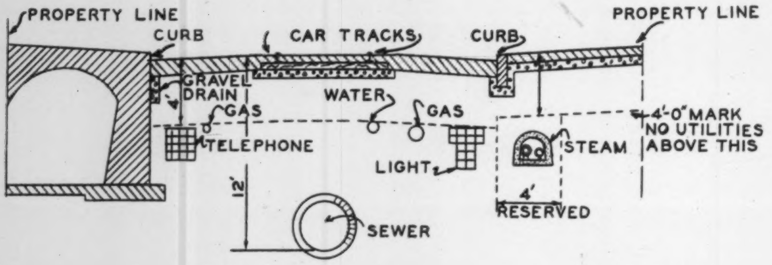


Fig. 1—Cross-Section of a Typical Street in Pittsburgh Showing Old Style Sidewalk Vault Extending to Curb Line, Brick Sewer, 4-in. Gas Line, 6-in. Water Line, 12-in. Gas Line, Steam and Return Pipes in Concrete Conduit, Telephone Company's Duct, and Light Company's Conduit

"Golden Triangle." The great majority of the streets within these limits are narrow, usually only 40 or 60 feet wide, including both sidewalks. Alley-ways are mostly 20 feet.

The earth beneath the roadways of these streets, for a number of feet down from the surface, is already well filled with pipes and conduits of other utility companies and of the city utility bureaus. Fig. 1 shows a cross-section of a typical street. Because of this congestion, the city department of public works has set aside a space four feet wide beneath the sidewalks, measured from the curb toward the buildings, where this space is available, for the laying of steam conduits.

A city ordinance provides that the top of the conduit be kept at least four feet beneath the surface of the sidewalk or roadway. This allows water and gas service lines to be run into buildings above the conduit, and sewer lines to pass from the buildings below the conduit. A typical conduit in the reserved location in the street is shown in Fig. 2.

The bureau of water of the city requires that steam conduits be laid not closer than one foot to any water line. When a water main lies in such position as to pass through a new steam manhole, either it is relocated or special sleeves are placed around it in the concrete manhole walls, during construction, to permit the removal of the water line later without chopping the walls.

Other Rules Affecting Location

Steam conduits for the larger size lines are made of concrete. The outside width of these structures, of course, varies with the size of lines to be encased, and the thickness of the walls found necessary for structural strength. Most of the conduits built are of less width than the four-foot reservation. The position that the structure will take therein must then be selected. In so doing the following conditions are considered: (Refer to Fig. 2 and the sides of the steam line area (a) next to the curb and (b) towards the buildings.)

When the trenching method of excavation is to be used, it is desirable to keep the conduit back from side (a) fifteen inches in order not to disturb the curb or gravel curb drain during construction.

Where there is a possibility that a building may be built, having a cellar vault with a wall extending to the edge of the reservation, as shown at (b), the structure should be built in the right of way in such position as to allow at least one foot of clear space for the driving of piling and for timbering when the wall is built.

When a wall of the kind mentioned above already exists, the conduit may be placed against it, saving timbering.

When conduit is laid deep and excavation is done by tunneling, the curb and curb drain will not be disturbed, if the structure is built out underneath them to the curb line (a). If desired, the conduit also may be built in close to side (b) if this method of excavation is followed. Only space for timbering need be left between the cellar vault wall and the conduit wall.

In alleys, and occasionally in streets, conditions necessitate the construction of steam line conduits underneath the roadway, instead of in the reserved sidewalk area. In this event care is taken to place the structure as far as possible from any underground electric cable conduit and manholes of the light company or of the telephone company to prevent the transmission of heat to the cables. The building of lines underneath and parallel to car tracks or other buried utility lines, except when laid very deep by tunneling, is avoided to save the extra expense of this class of construction.

When a steam conduit is to be laid so as to cross under a large brick sewer, it is necessary to figure on having at least one foot of space between the bottom of the sewer and the top of the steam structure for timbers to properly support the former during construction.

An effort is made to have all high pressure steam mains designed for 250 pounds per square inch located so that they will be straight between manholes in order to simplify the problem of expansion and line anchorage. Where possible the line is sloped down in the direction of the flow of steam.

The problem would be simpler if the second main (Fig. 2) carrying back the condensate were not also to be gotten into the conduit.

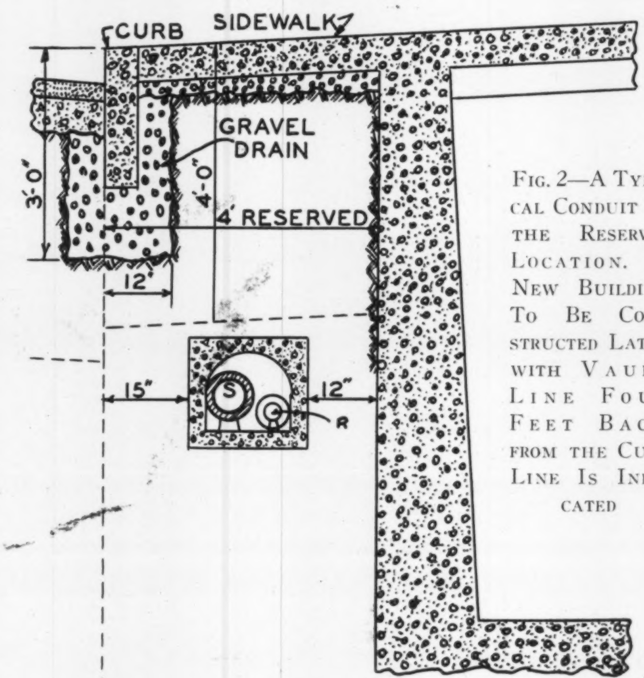


Fig. 2—A Typical Conduit in the Reserved Location. A New Building To Be Constructed Later with Vault Line Four Feet Back from the Curb Line Is Indicated

The Vault Problem

Permits were formerly given, when new buildings were being built, to extend the cellars out beneath the sidewalk so that the outside edge of the outer retaining wall coincided with the side of the curb next to the roadway, line (a) in Fig. 2. This wall then occupied approximately what is now the outer two feet of the steam line right-of-way. These vaults often were used to stock coal. In a considerable number of these the light company has constructed electric vaults. The other uses that are made of these sidewalk cellars or vaults are many. Some are well finished rooms. Others have hatchways with lifts by which much of the goods used in the buildings are conveyed to and from the basement. These old vaults are frequently encountered in planning line construction. Considerable diplomacy is often required in dealing with the tenants and owners of the abutting property to get their approval. The occupation of half of the allotted area by the retaining wall often necessitates placing one line above the other along the inside of the wall instead of side by side, as is usually done. See Fig. 3. Sometimes the line can be placed nearer the building than the allotted area, and at other times it is advisable to excavate and place the line out under the roadway. Any advantage gained by using the vault space in avoiding excavation, conduit construction or repaving may easily be lost if lifts or other machinery, shelving or other obstructions have to be moved to get the lines in place. On the other hand, the construction of mains through long unobstructed vaults is obviously less costly than placing lines of the same size in buried conduit.

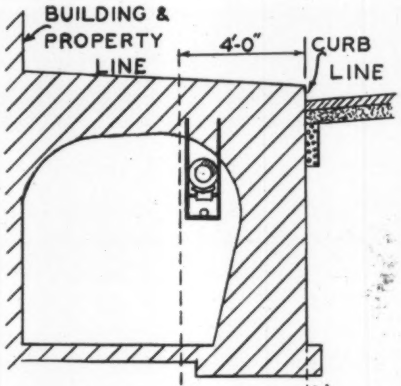
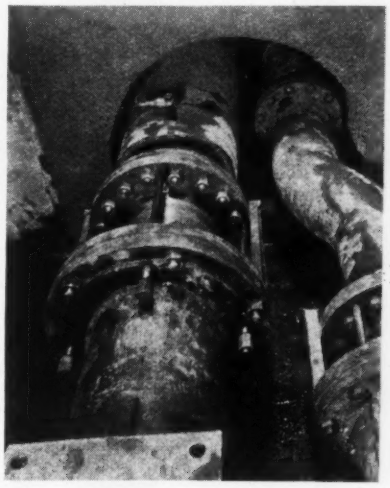


Fig. 3 — Condensate Return Line Under the Steam Line in Old Sidewalk Vault with Wall at Curb Line

On new building permits the city now ordinarily requires that cellar vault retaining walls be kept back toward the building so as not to encroach on the four foot right-of-way for the steam lines. An exception is made where electric vaults are to be built beneath sidewalks less than eleven feet wide. These vaults have to be seven feet wide at least in order to place ventilating gratings and hatchways of sufficient size in the sidewalk to permit the removal of equipment.

When these vaults are found in the proposed path for the steam conduit, it is necessary to decide whether it is better to put the lines through the vault in a special conduit or run them around or under it. Sometimes a single division wall is built to separate and serve both the vault and the conduit. This saves space.



Manhole at Liberty and Ferry Streets.

Minor Obstructions

Among the minor obstructions found in the path of steam conduit when located in the four foot reservations are poles, various service lines and ducts, fire hydrants, curb drains of broken stone and catch basins. Some of these have already been mentioned.

Poles usually do not extend sufficiently beneath the surface to obstruct the conduit. There are few wooden ones remaining. Arrangements have been made to cut off and properly support the bases of the steel street railway poles, which also carry street lights and traffic signals. Poles occasionally are moved, especially when they interfere with proposed manholes. This is also often done with minor service lines and ducts when they are found in the way. Fire hydrants and catch basins necessitate the lowering of the conduit roof to depths of approximately 6 feet and 8 feet respectively. Curb drains obstruct only at manholes. Here they are either built around the structure or carried through it as shown in Fig. 4.

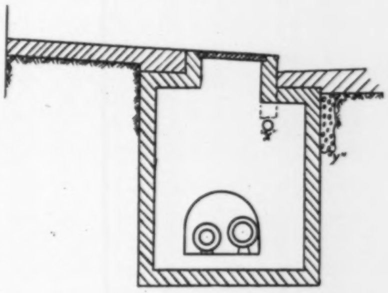


Fig. 4 — Showing the Curb Underdrainage Carried Past Man-Hole Either Through Pipe at *X* or Gravel at *Y*

Street Crossings

In designing a steam line to extend across a street from one block to the next it is, of course, necessary to find a space sufficiently large and suitable between the ducts and pipes running along the street to be crossed for the steam conduit to pass through. The depth at which the crossing can be made will have an influence on the distance down from the surface at which the conduit is laid, as it approaches and leaves the street crossing. This is especially true where a high pressure (250 lb.) steam line is to be laid and is to be a tangent, without curve or change in alignment or slope between man-holes.

In the street cross-section, Fig. 1, the gas and water lines and the tops of the conduits are shown at a depth of 4 feet below the surface, which is the minimum. The invert of the sewer is twelve feet below the roadway surface.

The problem is to find space underneath the pipes and ducts and over the top of the sewer for the steam conduit. The duct tops often are found considerably below the four foot depth and their bottoms down near the level of the top of the sewer. Where the lines cannot be laid above the sewer they must be placed under it, and a very deep laid steam conduit results.

A peculiar condition exists beneath some of the streets which were formerly flooded during periods of high water in the adjacent rivers. These streets were raised a number of years ago, but the old paving was allowed to remain in place. Consequently excavators run across additional paving a few feet below the surface. Water lines in place before the raising occurred are found deeper than usual. In another part of the district, steep grades have been removed from the streets and here occasionally a line is found nearer the surface than normally.

Some of the district lies on top of a shelf of very hard blue rock, which has to be cut out by air hammers. For the most part the excavation is in damp, sandy loam. It is conceivable that the kind of excavation to be encountered might determine where a line should go, as the removal of this rock is, of course, very much more expensive than the digging of earth.

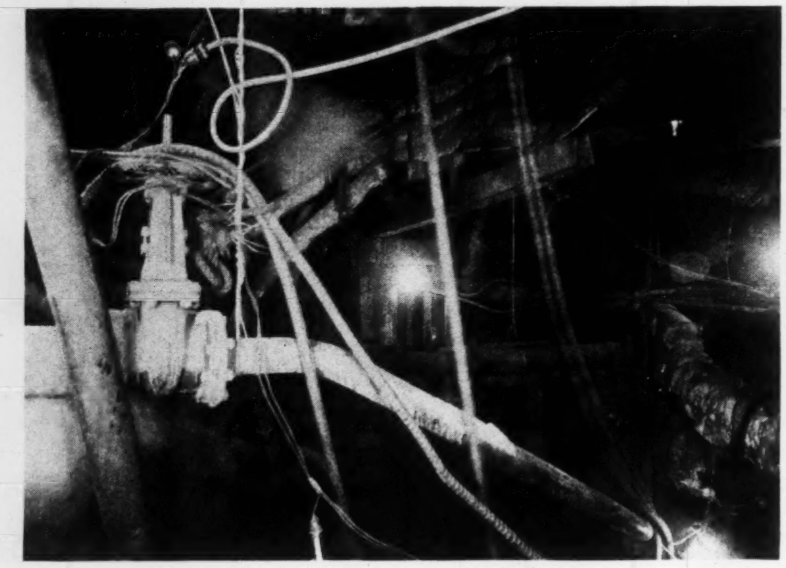
There follows a statement of the methods used in selecting locations for lines and conduits.

Preliminary Office Work

Before the actual survey in the street is begun, in order to avoid taking unnecessary notes, it is best to decide as well as possible what route would be best.

As a starting point some place on the existing steam system must be selected where there can be delivered sufficient additional steam at the desired pressure to supply the proposed new line. Perhaps the extension will not be loaded to capacity for a number of years, in which case it may be satisfactory if the existing line can supply it for that period. At the end of this time more steam must be delivered either by building an additional supply line or replacing the existing line with a larger one. This may easily be done if the line is in a steam tunnel of sufficient size or in a sidewalk vault or, with greater difficulty, if in a buried conduit. The difference between "steam tunnel" and "steam conduit" as here used is that the former is large enough for the lines to be accessible to the operating forces, while the latter is so small that the lines practically fill it and are inaccessible except at manholes.

If possible, the point selected on the line should be in a manhole or tunnel and should be anchored. Otherwise additional expense will be incurred in building a manhole and anchoring the line. This is necessary, for it is desirable to have accessible valves in the main line at branch connections and valves in the branches themselves. It is desirable also that there be no motion in the main where the branch takes off.



Penn Ave. Steam Tunnel—Vault C

Secondly, the point where the new lines will terminate must be decided upon. This usually is governed by two considerations, the wishes of a customer as to the place he desires the service to enter his premises and company policy and plans for extensions.

Finally, the proposed path of the lines between these termini must be determined upon. The general plan of the company for line extension is the first consideration. The second is the location and size of the buildings en route, either existing or proposed, and the third and final one is the length of line it will be necessary to build.

As explained above, generally the lines should be run close to the points where large demands are expected. The long branches will then be of smaller size and the total cost less. Consideration must be given to the effect of replacing old buildings with new ones, to the possible construction of future subways or other structures in the street and to any known difficulties that may be encountered on a route.

When more than one route is being considered, often all but one may be eliminated by making rough comparative estimates. This is done by the use of a chart based on experience, showing the cost of steam distribution lines per foot for various sizes complete, including a suitable return line, conduit, insulation and manholes.

Available Data

The next step is to look over all available data pertaining to the obstructions in the streets en route. This data consists of:

1. Steam company drawings of the existing lines and structures where the new line will begin.
2. The distribution system maps of the electric light company, the two gas companies, the telegraph company, the telephone company, the street railways company, the water department and the sewer department of the city.
3. Detail plans of light and telephone company manholes, sewer inlets, and occasionally details of sidewalk vaults of buildings.
4. A street map showing elevations.

The city rapid transit commission of Pittsburgh is preparing a composite set of plans showing all utility structures in most of the principal downtown streets, and the street elevations.

The data collected is used to make up a diagrammatic sketch such as is shown in Fig. 5. This is useful in the

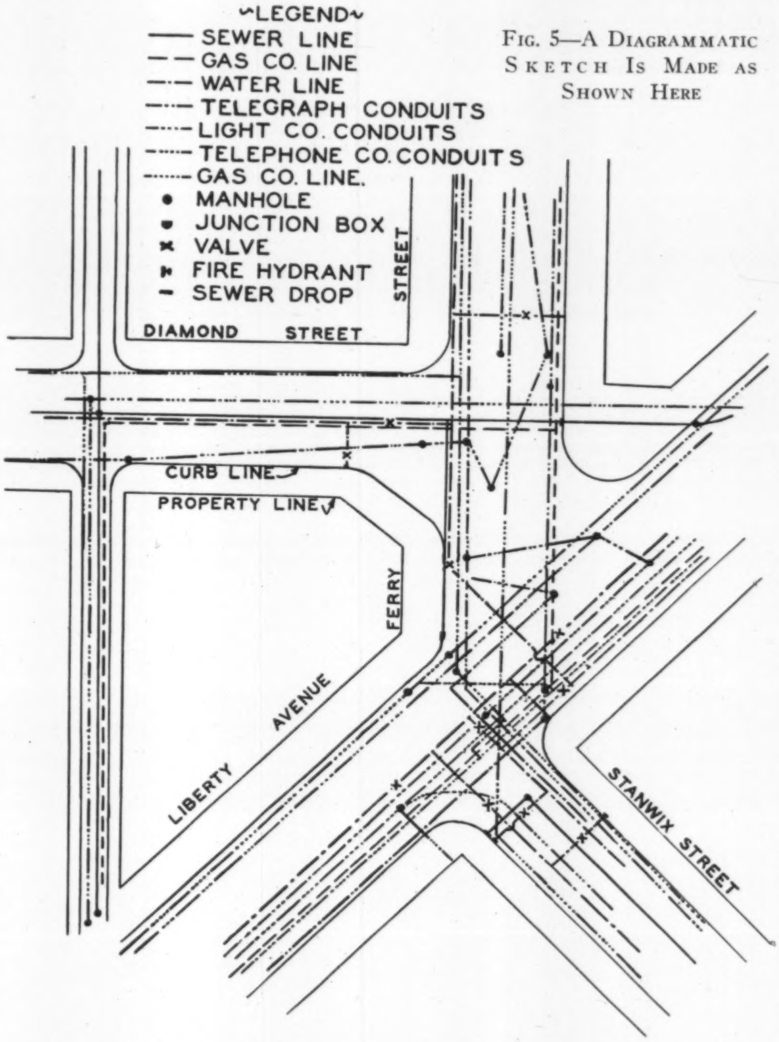


Fig. 5—A Diagrammatic Sketch Is Made as Shown Here

survey and later in the drafting room.

There is a coordination committee for public work in Pittsburgh consisting of representatives of the city, the boroughs, the county and the state governments and the various utilities. By following the minutes of the meetings of this committee it is possible to foresee any program affecting steam line extension.

On occasions the engineers of the other utilities and the city are contacted. The city utility engineer and the steam company's planning and operating engineers are frequently consulted and kept informed of all proposed extensions.

Surface Surveys

Actual measurement of the location of structures to be found in the streets is performed in two steps: surface measurements and sub-surface measurements.

The former often must be made at night or on Sundays when traffic is light. The crew consists of the survey engineer and one or two assistants. All manhole castings, valve boxes, poles, fire hydrants, car tracks, sewer inlets and all other structures indicating possible buried obstructions are located in respect to some point. Where the curb lines are straight they are taken as base lines and all measurements are made in respect to some point on them. This obviates the use of a transit, but this instrument will be found useful if the curbs are not straight. Actual elevations are taken with a level.

The diagrammatic sketch, Fig. 5, is important in visualizing the unseen conditions beneath the surface.

Sub-Surface Survey

The purpose of this part of the field work is to find the exact location of the obstructions previously plotted on the sketch, and any others to be found. The "unknowns" consist chiefly of private sewers not recorded on the sewer map and abandoned utility pipes.

Each manhole is entered and the locations of pipes and ducts passing through it are noted as well as the dimensions of the manhole itself if pertinent to the problem. All measurements are taken with respect to the center of the cover, which has been located already in the surface survey. In the same way the dimensions of curb inlets, sidewalk vaults, electric vaults and the pipes and ducts visible in them are measured.

Here a saving in labor can be effected if the surveyor takes only those dimensions that are required. This can not be emphasized too strongly, for the tendency is to take all dimensions, which is a waste of time, or to leave out some which will have to be secured later. Consider for example Fig. 6. "C" is a manhole with sides 1, 2, 3 and 4. A steam line is to be run from "A" to "B." It is obvious that dimensions of lines and ducts passing through wall 4 are vital and that those passing through walls 2 and 3 may be of interest, but that those passing through wall 1 are of no account. The eye should be quick to note any peculiarities of construction, such as a bend in a pipe and the angle and slope at which it passes through the wall. Electric ducts running between manholes will be found to slope towards either one or the other or both manholes. That is to say, the structure may be convex upward or it may slope to either manhole, but there will be no low spot which could hold drainage water.

Sometimes it is impossible to measure the depth of lines in valve boxes, due to the accumulation of debris.

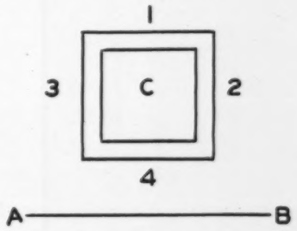
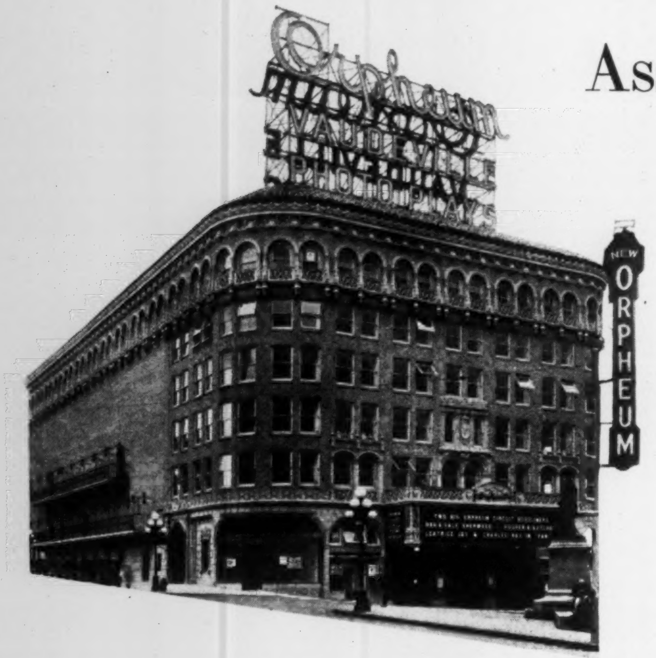


Fig. 6—The Surveyor Should Take Only Those Dimensions Which Will Be Used

In that case a measurement taken to the top of the valve stem (non-rising) will serve as well, as the distance from the center of the pipe to the top of the stem can be secured in catalogs.

The notes which are taken in the field are written on full letter-size sheets and must be neatly prepared and approved before being given to the draftsman. The sheets are loose leaf, so that the notes on each line extension can be filed in the general job folder with all other computations and data pertaining to it. The design of the lines is outside the scope of this article.

Two-Hour Readings of Conditions Assist in Operation of Theater Air Conditioning



THE ORPHEUM THEATER AND OFFICE BUILDING IN SEATTLE, WASHINGTON

By Erwin L. Weber*

WHEN the mechanical equipment for the Orpheum theater and office building in Seattle, Washington, was installed, the two were treated as separate buildings and all services were separately metered.

The office building was designed for an ultimate height of twelve stories, of which six have been built. Steam is purchased from the Puget Sound Electric Co. Originally the heating plant was designed as a two-pipe vacuum system. It was later re-designed as a two-pipe forced hot water system using a converter.

Water Temperatures

In this locality the prevailing winds are from the south and the cold winds from the north. The returns are, therefore, placed on two zones, one for the north facade and one for the south. Each branch extends to the return header through a gate valve and is provided with a thermometer. By throttling these valves, the average temperature of the water can be lowered on the leeward side and the heat forced to the windward side. The south return can also be throttled on sunny winter days to compensate for heating produced by solar radiation.

The following schedule of water temperatures has been found satisfactory:

Outdoor, in F	Average of water, in F
20	180
25	169
30	153
35	147
40	136
45	125
50	114
55	103
60	92

The average temperature is increased ½ deg. per mile of wind in excess of 10 miles per hour on the windward side. The circulating pump is arranged with a by-pass to permit gravity circulation during mild weather and after circulation is established.

Heating the Theater

In the theater portion, all direct radiation, coils, etc. are heated with steam on a two-pipe vacuum system. The stage is heated by direct radiation placed at three levels about 25 ft. apart to compensate for the local cold air drainage which would otherwise occur at the stage walls and create unpleasant drafts. Direct radiation is used in the executive offices and in the dressing rooms. Five wall radiators with a total of 182 sq. ft. of surface are placed on the basement ceiling, directly in back of the main entrance doors. These radiators are placed in a sheet metal housing and the slab above them is painted dead black to absorb heat and warm the floor at this point, thus checking cold drafts passing under the doors. The four stores on the first floor have separate returns for metering purposes.

Cooling the Air

During construction a well was drilled with a view of obtaining water at 52 F for cooling purposes, but no water-bearing gravel bed was encountered and it became necessary to install a refrigerating plant.

It was decided to use an upward system.

Many theaters have downward ventilation and practice recirculation to economize on refrigeration. In Seattle the wet bulb temperature seldom exceeds 62 F. The plant is normally operated so that the air is conditioned to 68½ dry bulb, 59½ deg. wet bulb, giving 64 deg. effective temperature at 60 per cent relative humidity. From this it will be seen that at 30 cu. ft. of air per minute per capita, the heat and moisture given off by the occupants would raise the wet bulb temperature of the vitiated air to 64 deg., which is higher than that found outdoors. Hence recirculation would defeat its purpose. In hot weather (62 deg. wet bulb) the incoming air is cooled to about 57 deg. wet bulb and heated to 68 deg. dry bulb, which gives an effective temperature of 63 deg. at 50 per cent relative humidity.

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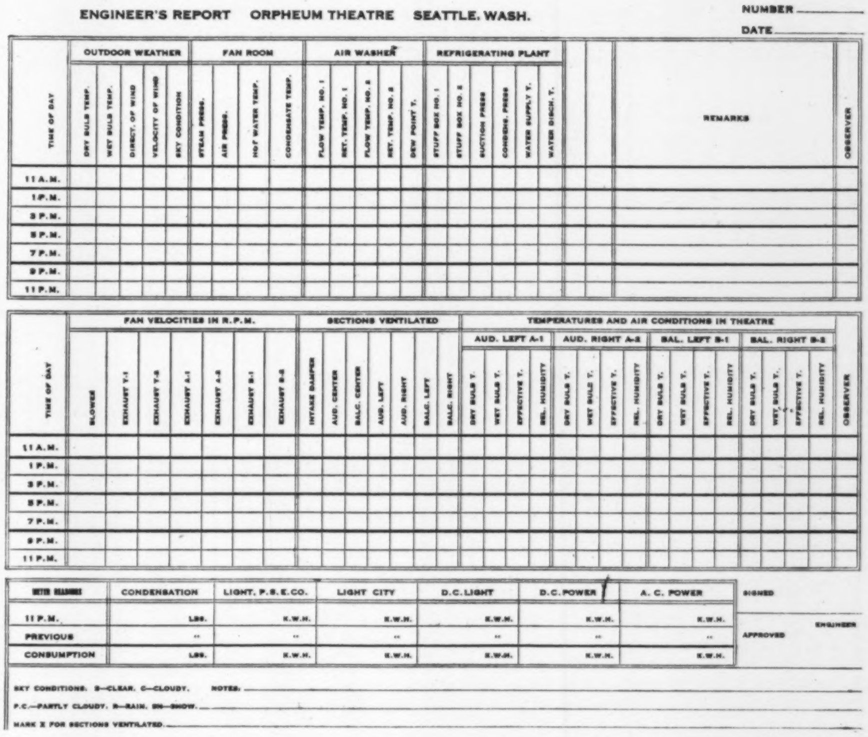
* Consulting engineer, Seattle, Wash.

The Air Washer

Air enters through the intake which is located about 20 ft. above the ground and passes to the air washer in the basement. The air washer has two sets of atomizing nozzles and one set of flood nozzles.



ENGINEER'S REPORT ORPHEUM THEATRE SEATTLE, WASH.

NUMBER ____

DATE ____

TIME OF DAY	OUTDOOR WEATHER					FAN ROOM				AIR WASHER					REFRIGERATING PLANT								REMARKS	OBSERVER
	DRY BULB TEMP.	WET BULB TEMP.	DIRECT. OF WIND	VELOCITY OF WIND	SKY CONDITION	STEAM PRESS.	AIR PRESS.	HOT WATER TEMP.	CONDENSATE TEMP.	FLOW TEMP. NO. 1	RET. TEMP. NO. 1	FLOW TEMP. NO. 2	RET. TEMP. NO. 2	DEW POINT T.	STUFF BOX NO. 1	STUFF BOX NO. 2	SUCTION PRESS	CONDENS. PRESS	WATER SUPPLY T.	WATER DISCH. T.				
11 A.M.																								
1 P.M.																								
3 P.M.																								
5 P.M.																								
7 P.M.																								
9 P.M.																								
11 P.M.																								

TIME OF DAY	FAN VELOCITIES IN R.P.M.							SECTIONS VENTILATED							TEMPERATURES AND AIR CONDITIONS IN THEATRE																OBSERVER
															AUD. LEFT A-1				AUD. RIGHT A-2				BAL. LEFT B-1				BAL. RIGHT B-2				
	BLOWER	EXHAUST T-1	EXHAUST T-2	EXHAUST A-1	EXHAUST A-2	EXHAUST B-1	EXHAUST B-2	INTAKE DAMPER	AUD. CENTER	BALC. CENTER	AUD. LEFT	AUD. RIGHT	BALC. LEFT	BALC. RIGHT	DRY BULB T.	WET BULB T.	EFFECTIVE T.	REL. HUMIDITY	DRY BULB T.	WET BULB T.	EFFECTIVE T.	REL. HUMIDITY	DRY BULB T.	WET BULB T.	EFFECTIVE T.	REL. HUMIDITY	DRY BULB T.	WET BULB T.	EFFECTIVE T.	REL. HUMIDITY	
11 A.M.																															
1 P.M.																															
3 P.M.																															
5 P.M.																															
7 P.M.																															
9 P.M.																															
11 P.M.																															

METER READINGS	CONDENSATION	LIGHT, P.S.E.CO.	LIGHT CITY	D.C. LIGHT	D.C. POWER	A. C. POWER
11 P.M.	LBS.	K.W.H.	K.W.H.	K.W.H.	K.W.H.	K.W.H.
PREVIOUS	"	"	"	"	"	"
CONSUMPTION	LBS.	K.W.H.	K.W.H.	K.W.H.	K.W.H.	K.W.H.

SIGNED ____ ENGINEER

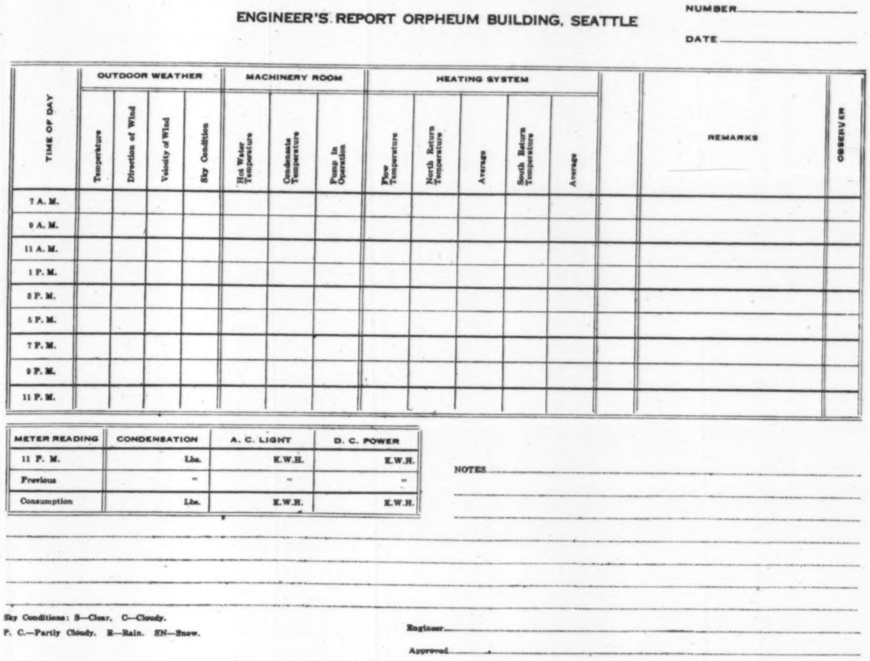
APPROVED ____

SKY CONDITIONS. S—CLEAR. C—CLOUDY. NOTES. ____

P.C.—PARTLY CLOUDY. R—RAIN. SN—SNOW. ____

MARK X FOR SECTIONS VENTILATED. ____

DURING THE OPERATION OF THE PLANT, THE ENGINEER INSPECTS IT AND RECORDS ALL VITAL INFORMATION. THESE DATA HAVE BEEN FOUND EXCEEDINGLY HELPFUL IN ASSURING ECONOMIC OPERATION AND LOCATING ANY FAULTS OR APPROACHING FAULTS. SPECIAL FORMS ARE PROVIDED FOR THE RECORDING OF THE INFORMATION OBTAINED



ENGINEER'S REPORT ORPHEUM BUILDING, SEATTLE

NUMBER ____

DATE ____

TIME OF DAY	OUTDOOR WEATHER				MACHINERY ROOM			HEATING SYSTEM						REMARKS	OBSERVER
	Temperature	Direction of Wind	Velocity of Wind	Sky Condition	Hot Water Temperature	Condensate Temperature	Pump in Operation	Flow Temperature	North Return Temperature	Average	South Return Temperature	Average			
7 A. M.															
9 A. M.															
11 A. M.															
1 P. M.															
3 P. M.															
5 P. M.															
7 P. M.															
9 P. M.															
11 P. M.															

METER READING	CONDENSATION	A. C. LIGHT	D. C. POWER
11 P. M.	Lbs.	K.W.H.	K.W.H.
Previous	"	"	"
Consumption	Lbs.	K.W.H.	K.W.H.

NOTES ____

Sky Conditions: S—Clear, C—Cloudy.
P. C.—Partly Cloudy. R—Rain. SN—Snow.

Engineer ____

Approved ____

There are no tempering coils, but the air is tempered in the air washer.

As steam is purchased and passes through a condensation meter to the sewer, it is desirable to extract all possible heat therefrom. The washer tank is divided into two compartments and two pumps are used, one for each set of atomizing nozzles. The water for the first set of atomizing nozzles is warmed by passing through a condensation converter, which consists of a 42 in. by 84 in. tank with two sets of copper coils each containing 50 sq. ft. of heating surface. The condensate from the entire system is discharged from the vacuum pump to the meter by first passing through the coil of the condensation tank and then through the lower coil of the condensation converter. There is usually sufficient heat thus extracted from the condensate to warm the air from 6 to 8 degrees in the first stage.

As the water in this first bank of nozzles may freeze when the outdoor temperature is below the freezing point, a steam line with a thermostat set at 35 deg. is provided for the upper coil of the condensation converter. With this arrangement condensate has at times been delivered to the sewer at a temperature as low as 45 degrees. The condensation tank is used as a preheater for the domestic hot water. During the summer the condensate passes through the condensation tank only. A thermostatically controlled 500 gal. storage tank with copper element is provided for the domestic hot water, in addition to the condensation tank.

Second Bank of Nozzles

The water for the second bank of atomizing nozzles is warmed by passing through a converter which has copper coils with 80 sq. ft. of heating surface and will heat 600 gallons of water per minute from 56 to 75 F, with steam at atmospheric pressure. Steam passes to it through a 5 in. diaphragm valve controlled by a thermostat in the blower discharge and set at 53 F (dew point control.) A second, or pilot, thermostat is located in the converter discharge line and set to close the diaphragm valve if the temperature of the water rises above 78 deg.

This provision is made so that if the regular thermostat fails to function the second thermostat will make it impossible for the temperature of the water to rise too high and thus fill the theater with vapor.

The refrigerating plant is provided with a 150-ton carbon dioxide compressor driven by a synchronous motor. The condenser was located in the surge tank room and the air washer tank was dropped below the floor and also raised in height to accommodate the expansion coils. These are of the trombone type provided with flood troughs with perforated drip gutters over the coils. Auxiliary ball cocks provide a low water level for summer use, thus securing a baudelot effect on the coils. This tank type of installation has proved satisfactory.

Air Introduction

The blower is direct-connected with a 220 volt d.c. motor of 30 hp. capacity, operates at 88 rpm normal speed and delivers 90,000 cu. ft. of air per minute. At an increase of speed to 105 rpm on field control the unit can deliver about 120,000 cu. ft. of air per minute. The motor weighs over 9,000 lb. and is one of the largest 30 hp. motors ever built.

The first set of reheating coils is thermostatically controlled to raise the temperature of the air to 63 F and discharges into a plenum chamber. The second set of reheating coils consists of four groups, each under separate thermostatic control; two discharge to the plenum space under the auditorium and two to the balcony.

Ducts *HAR* and *HAL,* to the right and left sides of the auditorium, and ducts *H*-10 and *H*-9, to the right and left sides of the balcony, are provided with remote controlled dampers, which can be closed when only the center of the house is occupied, thus economizing on power and steam.

The air enters the auditorium through outlets placed under every alternate seat and through lock registers and outlets in the balcony.

In place of feeding the air to the balcony from the plenum space under it, which is often dirty with plaster and leaks heavily, continuous airways to the outlets are provided. Metal chambers with two sides and ends are fixed under the balcony steps and between the radial girders. These chambers are fed by intersecting branch ducts supplied by risers from the basement.

Air heated through three banks of reheating coils, is discharged at the sides of the main entrance doors to mix with the cold air blowing into the entrance, which lies in the direction of the prevailing winds. These coils are individually controlled by switches located at the entrance and operated by the doorman, who can thus regulate the intensity of heat as required. A separate supply duct with three coils is also provided for the animal room. The coils are individually controlled by switches located on the stage just outside of this room. This provision is necessary as monkeys and other tropical animals require high temperatures.

An exceptionally large amount of fresh air is blown into the foyer to avoid any possibility of being met by the odors of vitiated air upon entering the theater and partly to counteract incoming drafts at the entrance.

The vitiated air from the auditorium passes through grilled ceilings to the attic space. Two exhaust fan rooms are located in the attic and insulated ducts lead from the attic space to these rooms. Each fan room has a double-intake, double-width exhaust fan, discharging up to 25,000 cu. ft. of air per minute, for the main ventilating units.

Exhaust fans *AR* and *AL* are connected with plaster grilles in the cove under the balcony and exhaust the air from the rear of the auditorium. Each fan is capable of exhausting up to 6,300 cu. ft. of air per minute.

Exhaust fans *TR* and *TL* are for toilets, retiring rooms, etc. The above exhaust fans are all provided with variable speed motors.

A separate exhaust fan is provided for the toilets in the office building, for the dressing rooms, for the operator's booth and for the animal room.

It has been found during operation that when the house is full, effective temperatures as low as 60 to 61 degrees are desirable, due to the inter-radiation of bodily heat among the closely concentrated audience. This necessitates adjusting the thermostats as the audience increases.

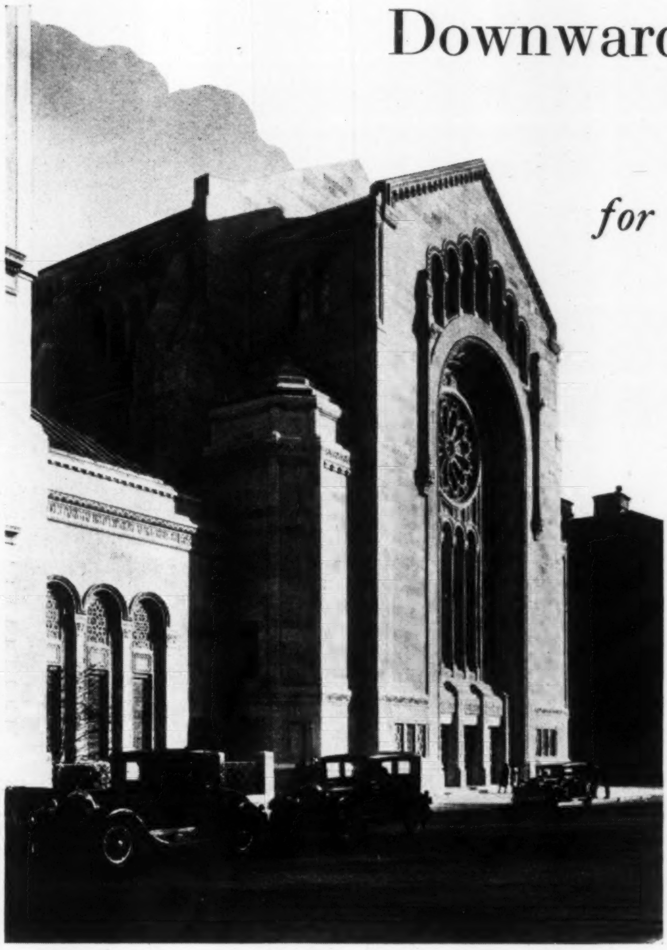
A signal system with drop annunciators notifies the engineer when the sides of the auditorium and balcony become filled, so that he can vary the speeds of the fans, open the remote controlled dampers and adjust the thermostats as required.

During operation the engineer inspects the entire plant every two hours, making observations and entering them on an *engineer's report sheet.* These observations include the outdoor weather conditions, the temperatures and pressures at various vital points in the heating and refrigerating plant, the speed of all fans, together with the dry bulb, wet bulb and effective temperatures and relative humidity for the right auditorium, left auditorium, right balcony and left balcony. Daily readings of the light, power and condensation meters and the consumption are also recorded.

VODEVILLE
IN SOUND

These observations and records have proved of inestimable value in training the engineer in the technique of the plant's operation and the essentials of air conditioning. They assist materially in locating any faults or approaching faults in the equipment and assure economic operation.

The mechanical equipment was installed by McNeal-Taylor Co. of Seattle. B. Marcus Priteca and Frederick J. Peters, Seattle were the architects for the building and the electrical, mechanical and sanitary equipment was designed by and installed under the supervision of the author.



Downward Ventilation in Winter, Upward in Summer, *for* Temple Emanu-El Conserves Fuel

By S. P. Moore

from an interview with

Alfred L. Jaros, Jr.

FIG. 1—GREAT ARCH AT GABLED ENTRANCE OF TEMPLE EMANU-EL IS ITS DISTINGUISHING ARCHITECTURAL KEYNOTE. ROBERT D. KOHN, CHARLES BUTLER, AND CLARENCE S. STEIN, ARCHITECTS

TEMPLE EMANU-EL, New York City, dedicated January 10, 1930, is the largest edifice in the world devoted to Jewish worship. Robert D. Kohn, Charles Butler, and Clarence S. Stein, New York City, were the associate architects on the project, with Francis L. Mayers, O. H. Murray, and Hardie Phillip as consulting architects. Jaros & Baum were the engineers on design, with Myron S. Falk as consulting engineer. We are indebted to Robert D. Kohn, architect, and Alfred L. Jaros, Jr., engineer, on this project for the data offered in this discussion.

This building is of the complex type that has come into increasing use for religious purposes during recent years. It combines the religious features, proper, with those of a community house. Its engineering features reflect its multiple purposes. The Temple itself is just under 200 feet in length. It has a width of 77 feet between the piers, and it has a height of 103 feet to the under side of the ridge of the ceiling. It is situated at the northeast corner of Fifth Avenue and Sixty-fifth Street on an L-shaped plot, fronting 150 feet on Fifth Avenue and 253 feet on Sixty-fifth Street. The northerly 50 foot section is occupied by Beth-El Chapel, set back about twenty feet from the Fifth Avenue line. The Temple proper, 100 feet in width, has a depth, including the sanctuary, of about 200 feet.

The easterly part of the plot is occupied by a community house which is eight stories high, but surmounted in part by a tower which goes up four stories further. Architecturally, this tower rather takes the place of the typical steeple of a church building. Its actual purpose is to contain the elevator machinery, water tank, and other similar features for the community house.

A large banquet hall and kitchen are in the west part of the plot directly beneath the temple itself. They are so arranged as to serve large gatherings at banquets, and also for lecture room or meeting room in case of need.

The Community House

The community house contains the boiler room and other features for mechanical operation of the building group in the basement. There is an independent auditorium on the ground floor. Space above this is mainly devoted to a library, class rooms, club rooms, trustees' rooms, offices, etc.

This portion of the building is heated by a two-pipe vacuum system with exposed radiators. There is a simple system of mechanical exhaust for various toilets, etc. The class rooms have received independent treatment by means of unit ventilators of approved school type, with venting through wardrobe closets to corridors which communicate with stairs, elevators and toilets. Similar units are provided in the library. In all cases, the units and any radiators in the same rooms with them are controlled by individual room thermostats.

Heating the Temple

The temple is heated entirely by recirculating gravity indirect stacks, independent of the mechanical ventilation. There are two separate heating zones for the temple, one comprising the space on the ground level, and the other the space on the balcony level. In one case only, the rear of the large balcony on the west end (Fig.

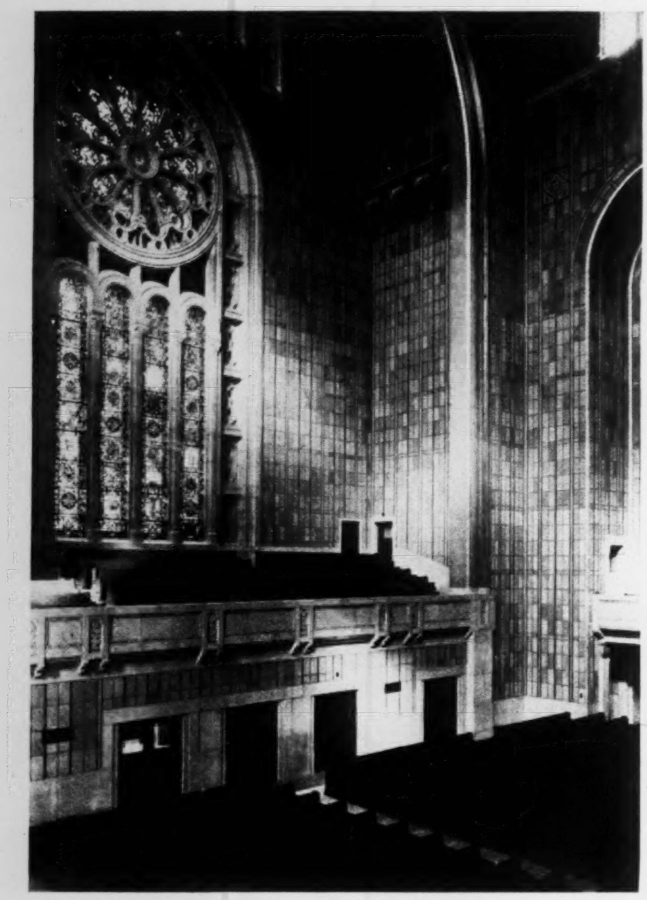


FIG. 2—WEST GALLERY AS SHOWN, FIVE GALLERIES ON EITHER SIDE, AND THE CHOIR LOFT ON THE EAST CONSTITUTE A SEPARATE HEATING ZONE

2) the indirect heating stacks are located in the construction of the balcony itself. In all other cases they are located in various pipe spaces and passages in the basement, and connected by galvanized-iron ducts and flues with the recirculating and warm-air grilles in the temple.

Typically, each bay of the temple, on each level, has its own independent heating stack, flues and grilles. This, in conjunction with individual thermostatic control, is intended to secure the best possible heat regulation in all parts of the temple. The roof space above the temple is heated by steam coils to prevent it from exercising a chilling effect upon those below.

The Heating Plant

The central heating plant for the entire group, located under the community house, includes three brick-set boilers, with special grates and electric driven, forced draft blowers for burning anthracite screenings, a vacuum return line pump, and all the usual appurtenances and accessories for such a plant. Steel plate boilers are used 5 ft. 6 in. in diameter by 16 ft. long. Each boiler contains 1,304 sq. ft. of heating surface made up of ninety-eight 3-in. tubes 16 ft. long, and of the boiler shells. These boilers are tested to carry 150 per cent of their normal rating without undue deterioration.

Forced draft is by means of two centrifugal fans each having capacity of 5,000 c.f.m. at 3 in. static pressure. The motors are 7½ hp., d-c., 230-volt, 1,050 r.p.m., 40 degree rise, compound wound, with combined armature and field-speed control. The system is fully sound-proofed and pressure is automatically regulated for each boiler.

Ventilation

The distinctive feature of this heating and ventilating plan is the versatility of use of many systems in different parts of the building. The four separate systems are major projects, each with special features. Three of them represent the best features of modern theater ventilation (apart from air conditioning) adapted to the structural and functional requirements of the modern church.

The banquet hall is served by a typical system of mechanical fresh air supply and exhaust. It is a slightly unusual feature here that the supply and exhaust fans are joined by a common base, carrying one motor whose shaft is coupled to both fans. Thus, the two fans are always run at the same speed and, whatever the speed at which the motor is run, the fresh air supply and exhaust are always in proper relation to each other.

The fresh air supply to this room is brought through registers located in both sides of the room. A few registers are located in the end walls. The location of these registers generally is just above head room. Exhaust is made through registers located at the break of the ceiling on the line of the inner columns; the raised central portion of the ceiling acts in effect as an exhaust hood for the room. Fig. 3 shows a small section of the banquet hall below and shows the relative location of the supply and exhaust ducts and registers. It also shows a part of the plenum ducts which connect with mushrooms in the floor of the temple.

The ventilating systems for the chapel, the temple, and the auditorium in the community house are radically different from the banquet hall system. In each of these three systems a single fan delivers to the room in question a mixture of fresh and recirculated air, and the suction at the inlet of same fan draws from the room the air which is to be recirculated.

Taking the temple system as the largest and most typical system of the three, the fan room is located in the seventh floor of the community house at a point adjacent to the roof space of the temple. In this particular case the outside air is brought down from the roof through an intake shaft, passed through air filters and tempering stacks of standard type, and brought to the inlet of the blower. The recirculated air is brought back through a duct to the blower inlet connection, where automatically-controlled pneumatic mixing dampers vary the proportion of all new and recirculated air to suit the weather and the number of people in the temple. The equipment is adjusted always to have at least one-third outside air, and a maximum of two-thirds when required.

The capacity of the blower is based upon moving 30 c.f.m. per person with all the seats and pews filled. From the blower the air goes to a large reversing damper of drum type, which makes it possible, by swinging one large damper, to reverse the entire operation of the ventilating duct system so as to provide at will either top supply and bottom exhaust, or bottom supply and top exhaust. The reversing damper is mounted on roller bearings, and is lined with felt along the edges.

From the reversing damper the ducts for the upper system run out directly through the roof space and connect to three rows of registers blowing directly down-

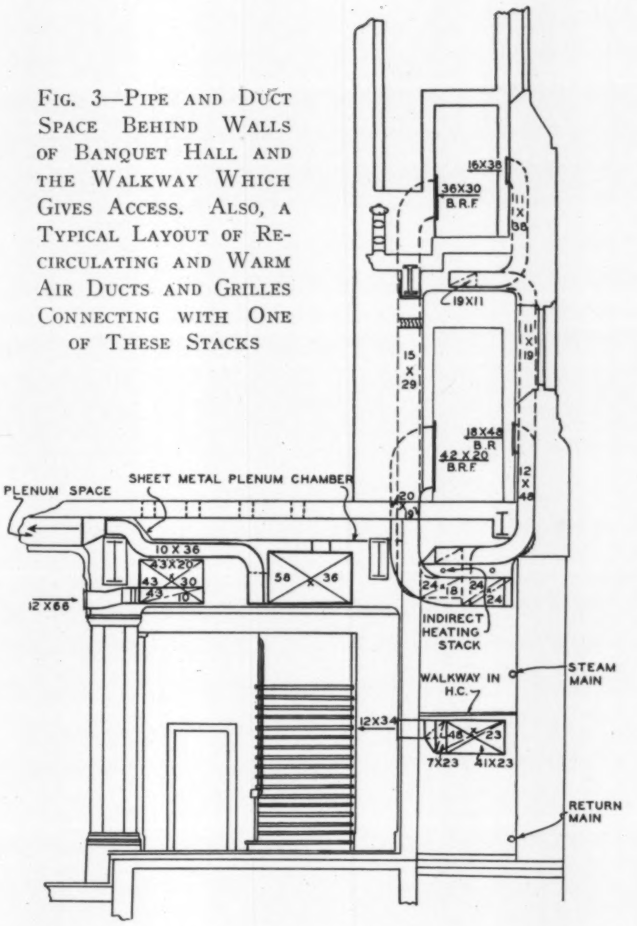


Fig. 3—Pipe and Duct Space Behind Walls of Banquet Hall and the Walkway Which Gives Access. Also, a Typical Layout of Recirculating and Warm Air Ducts and Grilles Connecting with One of These Stacks

ward, one on the center line and one on each side of the temple. These registers are designed for a velocity of about six hundred feet per minute because of their position from ninety to one hundred feet above the floor.

From the reversing drum the main ducts for this lower system connect with distributing ducts running over the sides of the banquet hall. These ducts then connect to a series of plenum spaces underneath the floor of the temple, each serving a group of mushroom ventilators located under the pews above. Branches from the same ducts are carried up to the floor of the rear balcony, where a plenum space is formed, serving similar mushrooms. There is nearly one mushroom for every two persons, thus giving sufficiently low velocities to avoid objectionable drafts at the feet of the occupants when the reversing damper is set for upward ventilation.

Operation During Cold Months

The intention is to use this system as an upward ventilating system with the air entering through the mushrooms and exhausted at the roof during the greater part of the year. The operation is to be reversed so as to run a downward ventilating system during the three or four coldest months, when the dryness of the indoor air might result in a chilly sensation if the air were introduced around the feet. During these colder months, operation as a down-draft system will tend to conserve fuel by resisting the tendency of the warm air to rise toward the roof.

When the system is operated as a downward ventilating system, the excess of the air blown in over and above that drawn back by the fan is expected to find its own way through doors, windows, or connecting passageways. The maintenance of a moderate positive pressure in this manner also assists the heating results by preventing indrafts at various doors and windows.

Operation During Warmer Months

During the warmer and greater portion of the year, when the equipment is run as an upward ventilating system, means are provided for getting rid of part of the surplus air, if found necessary, through auxiliary outlets in the temple roof. They are provided with two sets of louvre dampers, one operating in conjunction with the automatic mixing dampers, and the other operating by pneumatic push buttons and intended to remain tightly closed at all times when the system is run as a downward system.

The systems for the chapel and for the auditorium are similar in general scheme. The principal difference is that the chapel system is not provided with an air outlet duct, while the auditorium system has its air outlet taken directly from the reversing drum, so as to be used with either upward or downward ventilation.

The remainder of the ventilating systems consists of independent exhaust systems for the kitchen, the mezzanine toilets at the Fifth Avenue end, the mezzanine toilets at the community building end, and the toilets upstairs in the community building. There is also a small independent unit ventilator connecting to a number of supply grilles in the trustees' room on the eighth floor, and a separate fan exhaust for this room remotely controlled from the main fan room on the seventh floor.

The centrifugal fans required for the entire system are as follows:

System	CFM	S. P. Ins.	Ft. Min. P.V.	O.V.	R.P.M.
Chapel Supp.	10,500	1¼	3600	2000	425
Banquet Hall Supp.	20,000	1¼	3500	1800	300
Banquet Hall Exh.	16,000	¾	3100	2050	300
Kitchen Exh.	9,700	1½	3900	2200	520
W. Ct. Rm. Exh.	6,700	1	3300	1900	585
Auditor. Supp.	20,000	1¼	3500	1800	300
E. Coat Rm. Exh.	3,900	1	3300	1950	650
Forced Draft	5,000	3	5000	2500	1050
Forced Draft	5,000	3	5000	2500	1050
B'm't Proj. Rm. Exh.	500	⅜	3800	400	1150
Auditor. Proj. Rm. Exh.	500	⅜	3800	400	1150
Temple Supp.	72,000	1½	4000	2400	195
Toilet Exh.	5,000	¾	2900	1900	500
Trustees' Rm. Exh.	1,200	¼	2000	1350	588

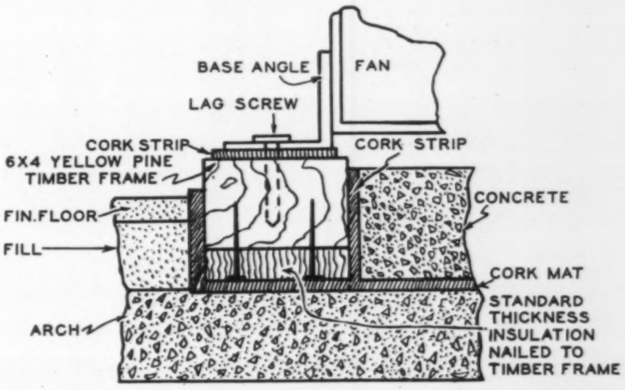


Fig. 4—Detail of Fan Frame and Support

Economical Steam Pipe Sizes *for* Heating, Process and Power Steam

Article No. 3

By A. W. Moulder

In the first two installments of this series on steam pipe sizes, tables were published giving the pounds of steam carried in pipe for drops per foot in pressure of .0001 to .0250. More of the tables appear in this issue; when the series is complete, data will have been published for drops up to one pound per foot.

For an explanation of the tables, refer to Article No. 1 in the August HEATING, PIPING AND AIR CONDITIONING.

EXAMPLE NO. 2—OFFICES, HOTELS, STORES, ETC.

We have just finished the working out of pipe sizes for a heating system in an industrial building where the heating units were relatively large and widely separated and have seen that it is practical in such cases to equalize fairly accurately the resistance to each individual unit. (Pp. 751-59, September, 1930, issue.)

We are now considering for our Example No. 2 (see Fig. 4) the type of heating system which is encountered in office or mercantile properties or hotels. Here we have a multiplicity of small radiator units, a great number of risers and main piping usually relatively short in total length.

Therefore, in this type of heating system, the saving in first cost cannot be an important factor because of the relatively small cost of the mains. There is not much to be saved in first cost by cutting down the size of risers from 2 in. to 1½ in. or even 1¼ in. A real saving in first cost can only be made in long horizontal mains. So whatever is done toward equalization in this system must be with a view to more efficient operation. That this is a desirable end is evidenced by the fact that we are purchasing more and more devices to give this result, for instance, systems in which orifices are used for equalization, adjustable radiator valves which can be set to allow only a certain maximum flow, zone systems, etc.

While the method described in this example only approximates perfection, it probably approaches it as closely as is usually the case. Since it is, in most cases, easily predetermined, literally costs somewhat less than nothing and in every way is entirely practical, why not have it to effect a saving in fuel bills?

It is not recommended that the engineer carry equalization to each radiator on the job, but rather that he assume that the drop in pressure through each individual riser will be negligible or at least practically the same if figured from any practical rule-of-thumb table of pipe sizes; then, treat the entrance of the first or "common" tee of each riser as though it were the entrance to a unit of radiation rather than a separate riser. In other words, every riser will be calculated as a unit and equalization to the various units in the riser ignored.

First step: Although the sizes of pipe in the individual risers and radiator connections will have no bearing on the problem of equalization, we will figure through the last riser just to establish the fact that with ordinary pipe sizes, the drop in pressure is negligible. It is for the engineer, in this instance, to use sizes which he has found practical for the type of system, whether one-pipe gravity, two-pipe gravity or vacuum steam.

Riser No. 9, Fig. 4

Sizes of pipe will be used for this riser or unit the same as would be found by using data given in the *A. S. H. V. E. Guide*, 1930 edition, page 358. Although in this example the total equivalent run is approximately 600 ft., we are going to use a somewhat higher total drop than 8 oz. (the basis of the table) and therefore will use sizes as shown under Col. D, which is for 300-ft. runs and will therefore produce somewhat more economical pipe sizes.

It will be seen from the following compilation that the resistance through the riser may be considered negligible in proportion to the total drop allowable and furthermore, any attempt to equalize connections of various radiators in the riser or unit would not be worth while. All risers and radiator connections should be sized on the same basis, regardless of which table of capacities the engineer may elect to use.

Load		Size, Inches	Length		Equiv. Length	Drop per Ft.	Drop in Piece	Total Drop
Sq. Ft.	Pounds		Ft.	Ells				
580	145	2	14	....	14	.0015	.0210	.0210
500	125	2	14	....	14	.0012	.0168	.0378
420	105	2	14	....	14	.0008	.0112	.0490
340	85	2	14	....	14	.0005	.0070	.0560
260	65	1½	14	....	14	.0013	.0182	.0742
180	45	1¼	14	....	14	.0015	.0210	.0952
100	25	1¼	18	....	18	.0005	.0090	.1042
100	25	1¼	4	4	24	.0005	.0120	.1162

Second step: Having arrived at practical sizes for

PIPE SIZES FOR TWO-PIPE VAPOR HEATING SYSTEMS, WHERE EQUIVALENT LENGTH OF RUN FROM BOILER OR SOURCE OF SUPPLY TO FARTHEST RADIATOR EXCEEDS 200 FT.

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Note (1): Radiator branches more than 8 ft. in length should be one size larger than shown in Column *I*. (2) This table is for systems which are open to atmosphere or operate under slight pressure or partial vacuum without use of vacuum pumps. (3) This table applies where pipes are properly reamed. No allowances for defective material or workmanship have been made. (4) Capacities based on ¼ lb. condensation per square foot per hour equivalent radiation and actual diameter of standard pipe. (5) Extra length to be added to straight run of pipe for various fittings and valves to determine equivalent length. (6) Mains are to be proportioned according to the equivalent length of run from the boiler or source of supply to the farthest radiators supplied by the main.

Capacity in Sq. Ft. of Equivalent Radiation

Based on 8 oz. Total Pressure Drop

Pipe Size In.	Equivalent Length of Pipe from Boiler to Farthest Radiator, Including Main and Riser. (See *Note 6.*) Supply Main Dripped and Branches to Risers Dripped—Steam and Condensate flowing in same direction.									Maximum Capacities	
	100 Ft.	200 Ft.	300 Ft.	400 Ft.	500 Ft.	600 Ft.	800 Ft.	1000 Ft.	1200 Ft.	Supply Risers Up-Feed	Branches to Supply Risers and Radiators Not Dripped
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*J*	*K*	*L*[1]
1	157	111	92	79	70	65	56	49	46	**56**	26
1¼	346	245	200	173	154	141	122	110	100	**122**	58
1½	538	380	310	269	240	220	190	165	155	**190**	95
2	1091	771	630	546	487	446	386	345	315	**386**	195
2½	1797	1270	1036	898	803	734	635	568	518	**635**	395
3	3289	2326	1896	1645	1470	1342	1163	1040	948	**1129**	700
3½	4913	3474	2838	2457	2196	2006	1737	1552	1419	**1548**	1150
4	6950	4914	4022	3475	3106	2828	2457	2196	2011	**2042**	1700
5	12,858	9092	7424	6429	5747	5250	4546	4062	3712	....	3150
6	21,105	14,924	12,168	10,553	9433	8618	7462	6669	6084	....	....
8	43,934	31,066	25,364	21,967	19,638	17,935	15,533	13,880	12,682	....	....
10	80,171	56,689	46,288	40,085	35,836	32,730	28,345	25,334	23,144	....	....
12	128,672	90,985	74,290	64,336	57,516	52,530	45,492	40,660	37,145	....	....
16	240,245	169,879	138,381	121,012	107,389	98,500	84,849	75,917	69,671	....	....

Pipe Size Inches		Return Mains and Risers									
Riser	Main	100 Ft.	200 Ft.	300 Ft.	400 Ft.	500 Ft.	600 Ft.	800 Ft.	1000 Ft.	1200 Ft.	
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	
....	¾	1130	800	653	568	505	462	400	358	326	**Different makes of supply and return valves, steam traps and other specialties vary as to capacity, therefore use size as recommended for any particular make. Vertical connection to be of same size as valve and trap used. Return horizontal runout to be not less than ¾ in.**
¾	1	1977	1400	1143	994	884	810	700	626	570	
1	1¼	3390	2400	1960	1704	1515	1387	1200	1073	976	
1¼	1½	5370	3800	3103	2696	2400	2195	1900	1698	1547	
1½	2	11,300	8000	6533	5680	5050	4622	4000	3575	3256	
2	2½	18,925	13,400	10,940	9510	8460	7745	6700	5990	5453	
2½	3	30,230	21,400	17,460	15,190	13,510	12,360	10,700	9565	8710	
3	3½	45,200	32,000	26,130	22,710	20,200	18,490	16,000	14,300	13,020	
3½	4	62,180	44,000	35,950	31,220	27,800	25,430	22,000	19,660	17,910	
4	5	109,300	77,400	63,200	54,920	48,800	44,720	38,700	34,600	31,500	
5	6	175,100	124,000	101,200	88,000	78,200	71,700	62,000	55,410	50,450	

all of the risers and radiator connections and having agreed that the drop through them may be assumed to be approximately the same through each riser from the common or first tee, the next step is to arrive at proper sizes for the mains and find the *base drop.* This operation is the same as for Example No. 1 (explained in the September issue) but is again recorded for convenience.

The total equivalent length is found to be approximately 600 ft. Assuming that we have an allowable total drop in pressure of 2 lb., the average drop in pressure per foot will be approximately .0033 per ft.

The following are the calculations for main sizes:

This shows a base drop from source of supply to common or first tee of farthest riser of 1.817 lb. Adding to this the drop through the riser and last connection the total theoretical drop in the system is 1.817 plus .116 = 1.933 lb.

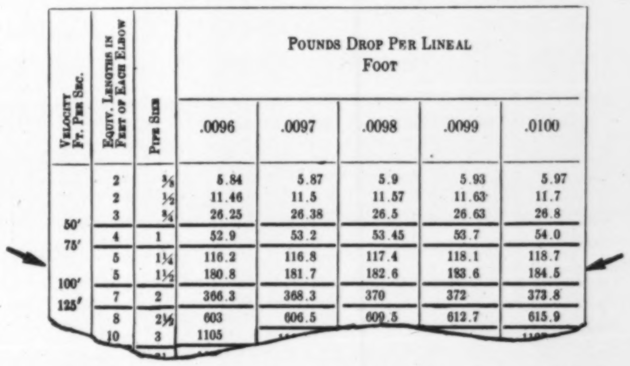
Piece No.	Load, Pounds	Length		Equiv. Length	Size, Inches	Drop per Ft.	Drop in Piece	Total Drop
		Ft.	Ells					
Main Riser	2690	140	6	242	5	.0037	.8954	.8954
A	1590	40	1	54	4	.0045	.2430	1.1384
B	1420	20	..	20	4	.0036	.0720	1.210
C	1250	20	..	20	4	.0028	.0560	1.266
D	1050	20	..	20	4	.0020	.0400	1.306
E	880	20	..	20	3½	.0028	.0560	1.362
F	710	30	1	40	3	.0040	.1600	1.522
G	590	20	..	20	3	.0028	.0560	1.578
H	420	20	..	20	2½	.0047	.0940	1.672
I	270	12	1	20	2½	.0020	.0400	1.712
J	100	...	..	...	...	.....	.....	.....
K	170	22	4	50	2	.0021	.1050	1.817

Third step: Having established the base drop, the next problem is to develop a drop in pressure equivalent to this base at the top of every riser or rather at the common or first tee of every riser.

The following represents the calculations to accomplish this for riser or unit (4):

Connection *P* (Fig. 4)
Load: 170 lb.
Length: 22′ and 5 ells.
Total equivalent: 47 ft.
Drop in main at connection: 1.306.
Base drop 1.817 — 1.306 = 0.511 drop required through connection.
0.511 ÷ 47 ft. = .010 average drop per ft. on basis of connection being all 1½″.
Referring to the table for a drop of .010 we find the 1½″ connection would pass 184.5 lb. of steam or 14.5 lb. more than required. This indicates the load of 170 lb. comes between 1¼″ and 1½″. In this case the

Pounds of Steam Carried in Pipe
for
Drops per Lineal Foot of .0096 to .0100 Lb.



Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0096	.0097	.0098	.0099	.0100
	2	⅜	5.84	5.87	5.9	5.93	5.97
	2	½	11.46	11.5	11.57	11.63	11.7
50′	3	¾	26.25	26.38	26.5	26.63	26.8
75′	4	1	52.9	53.2	53.45	53.7	54.0
	5	1¼	116.2	116.8	117.4	118.1	118.7
100′	5	1½	180.8	181.7	182.6	183.6	184.5
125′	7	2	366.3	368.3	370	372	373.8
	8	2½	603	606.5	609.5	612.7	615.9
	10	3	1105				

A 1½ IN. CONNECTION WOULD PASS 184.5 LB. OF STEAM WITH A DROP PER FOOT OF .0100.

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0255 to .0275 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0255	.0260	.0265	.0270	.0275
	2	⅜	9.525	9.625	9.72	9.81	9.89
75′	2	½	18.67	18.86	19.05	19.22	19.4
100′	3	¾	42.8	43.23	43.65	44.05	44.45
125′	4	1	86.2	87.1	87.95	88.75	89.6
	5	1¼	189.5	191.4	193.3	195.1	196.9
150′	5	1½	294.6	297.6	300.5	303.3	306
	7	2	597	603	609	614	620
	8	2½	983	993	1003	1012	1021
	10	3	1800	1818	1838	1853	1870
	12	3½	2690	2717	2742	2770	2795
	14	4	3801	3839	3877	3913	3948
	15	4½	5172.2	5224.1	5275.9	5324.5	5373.1
	17	5	7037	7108	7178	7244	7310
	20	6	11547	11663	11779	11887	11996
	24	7	17105	17276	17448	17609	17769
	27	8	24044	24285	24526	24752	24978
	30	9	32485.8	32811.5	33137.2	33442.5	33747.8
	34	10	43875.1	44315	44754.8	45167.2	45579.5
	40	12	70418.8	71124.7	71830.7	72492.5	73154.4
	47	14	91199.7	92114	93028.3	93885.4	94742.6
	53	16	131480.6	132798.7	134116.8	135352.5	136588.2
	60	18	180934.5	182748.4	184562.2	186262.8	187963.3
	67	20	240192.3	242600.2	245008.1	247265.6	249523
	81	24	390397	394310.7	398224.5	401893.6	405562.7

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0280 to .0300 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0280	.0285	.0290	.0295	.0300
	2	⅜	9.99	10.07	10.15	10.23	10.33
75′	2	½	19.57	19.75	19.9	20.1	20.27
100′	3	¾	44.7	45.26	45.66	46.05	46.44
125′	4	1	90.4	91.2	92.0	92.8	93.6
150′	5	1¼	198.6	200.5	202.2	203.9	205.7
	5	1½	308.8	311.6	314.4	316	319.8
	7	2	625.5	631	637	642	648
	8	2½	1030	1039	1048	1057	1066
	10	3	1886	1904	1922	1934	1953
	12	3½	2820	2845	2870	2895	2920
	14	4	3984	4020	4056	4089	4125
	15	4½	5421.7	5470.4	5519	5564.3	5613
	17	5	7376	7443	7509	7571	7637
	20	6	12104	12213	12321	12422	12531
	24	7	17930	18091	18252	18402	18562
	27	8	25204	25430	25656	25867	26093
	30	9	34053.1	34358.4	34663.8	34948.7	35254
	34	10	45991.9	46404.3	46816.6	47201.5	47613.8
	40	12	73816.2	74478	75139.8	75757.6	76419.4
	47	14	95599.7	96456.9	97314	98114	98971.1
	53	16	137823.9	139059.7	140295.4	141448.7	142684.4
	60	18	189663.8	191364.3	193064.8	194651.9	196352.5
	67	20	251780.5	254037.9	256295.4	258402.3	260659.8
	81	24	409231.9	412901	416570.2	419994.7	423663.9

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0305 to .0325 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0305	.0310	.0315	.0320	.0325
	2	⅜	10.42	10.5	10.59	10.68	10.75
75′	2	½	20.42	20.6	20.75	20.92	21.08
100′	3	¾	46.81	47.2	47.6	47.97	48.32
125′	4	1	94.3	95.0	95.9	96.65	97.3
150′	5	1¼	207.3	209	210.8	212.4	214
	5	1½	322.3	325	327.8	330.3	332.6
	7	2	653	658	664	669	674
	8	2½	1075	1084	1093	1102	1110
	10	3	1970	1985	2001	2020	2032
	12	3½	2943	2965	2990	3013	3035
	14	4	4158	4191	4227	4261	4291
	15	4½	5658.3	5703.7	5752.3	5797.7	5839.8
	17	5	7698	7760	7826	7888	7945
	20	6	12632	12734	12842	12943	13037
	24	7	18712	18862	19023	19173	19313
	27	8	26303	26514	26740	26951	27147
	30	9	35539	35824	36129.3	36414.2	36678.9
	34	10	47998.7	48383.6	48795.9	49180.8	49538.2
	40	12	77037.1	77654.8	78316.6	78934.3	79507.9
	47	14	99771.1	100571.1	101428.3	102228.3	102971.1
	53	16	143837.8	144991.1	146226.8	147380.2	148451.1
	60	18	197939.6	199526.8	201227.3	202814.4	204228.2
	67	20	262766.7	264873.7	267131.1	269238.1	271194.5
	81	24	427088.4	430512.9	434182.1	437606.6	440786.5

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0330 to .0350 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0330	.0335	.0340	.0345	.0350
	2	⅜	10.83	10.92	11.0	11.08	11.16
75′	2	½	21.25	21.4	21.6	21.7	21.9
100′	3	¾	48.7	49.08	49.45	49.8	50.12
125′	4	1	98.1	98.85	99.6	100.3	101
150′	5	1¼	215.7	217.3	219	220.5	222
	5	1½	335.2	338	340.3	342.7	345.2
	7	2	679	684	689.7	694.5	699.5
	8	2½	1118	1127	1135	1143	1152
	10	3	2050	2062	2080	2093	2110
	12	3½	3060	3085	3110	3130	3150
	14	4	4325	4358	4392	4422	4453
	15	4½	5885.2	5930.5	5975.9	6018.0	6060.2
	17	5	8007	8069	8130	8188	8245
	20	6	13139	13240	13341	13435	13529
	24	7	19463	19613	19763	19902	20041
	27	8	27360	27569	27780	27976	28171
	30	9	36963.8	37248.8	37533.7	37798.4	38063
	34	10	49923.1	50307.9	50692.8	51050.2	51407.6
	40	12	80125.6	80743.3	81361.1	81934.6	82508.2
	47	14	103771.1	104571.1	105371.1	106114	106856.8
	53	16	149604.5	150757.8	151911.1	152982.1	154053.1
	60	18	205875.3	207462.5	209049.6	210523.4	211997.2
	67	20	273301.5	275408.4	277515.4	279471.8	281428.3
	81	24	444211.1	447635.6	451060.1	454240.1	457420

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0355 to .0375 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0355	.0360	.0365	.0370	.0375
75′	2	⅜	11.25	11.32	11.4	11.47	11.55
	2	½	22.05	22.2	22.35	22.5	22.65
100′	3	¾	50.5	50.65	50.97	51.57	51.9
125′	4	1	101.7	102.5	103.1	103.8	104.6
150′	5	1¼	223.7	225.3	226.8	228.3	230
	5	1½	347.8	350.2	352.6	355	357.4
	7	2	704.5	709	714	719	724
	8	2½	1160	1168	1176	1184	1192
	10	3	2125	2140	2155	2170	2185
	12	3½	3178	3195	3220	3240	3260
	14	4	4487	4518	4549	4580	4611
	15	4½	6105.5	6147.7	6189.8	6231.9	6274.1
	17	5	8307	8364	8421	8479	8536
	20	6	13631	13725	13819	13913	14007
	24	7	20191	20331	20470	20609	20749
	27	8	28382	28578	28774	28970	29166
	30	9	38347.9	38612.5	38877.2	39141.8	39406.4
	34	10	51792.4	52149.8	52507.2	52864.6	53221.9
	40	12	83125.9	83699.5	84273.1	84846.7	85420.3
	47	14	107656.8	108399.7	109142.5	109885.4	110628.3
	53	16	155206.4	156277.4	157348.3	158419.3	159490.2
	60	18	213584.3	215058.1	216531.9	218005.7	219479.4
	67	20	283535.2	285491.7	287448.1	289404.6	291361
	81	24	460844.5	464024.5	467204.4	470384.3	473564.2

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0380 to .0400 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0380	.0385	.0390	.0395	.0400
75′	2	⅜	11.63	11.7	11.78	11.85	11.93
100′	2	½	22.8	22.95	23.10	23.23	23.4
	3	¾	52.27	52.6	52.92	53.38	53.63
125′	4	1	105.2	106	106.6	107.3	108
150′	5	1¼	231.5	233	234.4	236	237.5
	5	1½	359.8	362.2	364.3	367	369.2
	7	2	729	734	738	743	748
	8	2½	1200	1208	1215	1224	1232
	10	3	2200	2212	2228	2240	2255
	12	3½	3285	3310	3330	3350	3370
	14	4	4642	4673	4701	4732	4763
	15	4½	6316.2	6358.3	6397.2	6439.3	6481.5
	17	5	8593	8651	8704	8761	8818
	20	6	14101	14195	14264	14358	14470
	24	7	20888	21027	21156	21295	21435
	27	8	29362	29557	29738	29934	30130
	30	9	39671	39935.6	40179.8	40444.5	40709.1
	34	10	53579.3	53936.7	54266.6	54624	54981.4
	40	12	85993.9	86567.5	87096.9	87670.5	88244.1
	47	14	111371.1	112114	112799.7	113542.5	114285.4
	53	16	160561.2	161632.1	162620.7	163691.7	164762.6
	60	18	220953.2	222427	223787.4	225261.2	226735
	67	20	293317.5	295273.9	297079.9	299036.4	300992.8
	81	24	476744.2	479924.1	482859.4	486039.3	489219.3

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0405 to .0425 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0405	.0410	.0415	.0420	.0425
75′	2	⅜	12.0	12.09	12.15	12.23	12.3
100′	2	½	23.5	23.7	23.8	24.0	24.13
125′	3	¾	53.93	54.3	54.63	54.93	55.27
150′	4	1	108.6	109.3	110	110.6	111.3
	5	1¼	239	240.4	241.9	243.3	244.8
	5	1½	371.5	374	376	378.3	380.5
	7	2	752.5	757	761	766	771
	8	2½	1238	1247	1254	1262	1269
	10	3	2270	2283	2297	2310	2325
	12	3½	3390	3410	3430	3450	3470
	14	4	4792	4823	4851	4880	4908
	15	4½	6520.4	6562.5	6601.4	6640	6679
	17	5	8871	8928	8981	9034	9087
	20	6	14557	14651	14738	14824	14911
	24	7	21563	21703	21831	21960	22088
	27	8	30311	30507	30687	30868	31049
	30	9	40953.3	41217.9	41462.2	41706.4	41950.7
	34	10	55311.2	55668.6	55998.5	56328.4	56658.3
	40	12	88773.6	89347.2	89876.6	90406.1	90935.5
	47	14	114971.1	115714	116399.7	117085.4	117771.1
	53	16	165751.2	166822.2	167810.7	168799.3	169787.9
	60	18	228095.4	229569.1	230929.5	232290	233650.4
	67	20	302798.8	304755.2	306561.2	308367.1	310173.1
	81	24	492154.6	495334.5	498269.8	501205.1	504140.4

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0430 to .0450 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0430	.0435	.0440	.0445	.0450
75′	2	⅜	12.38	12.45	12.5	12.53	12.65
100′	2	½	24.25	24.4	24.52	24.6	24.8
125′	3	¾	55.6	55.9	56.25	56.55	56.85
150′	4	1	112	112.6	113.3	113.85	114.5
	5	1¼	246.3	247.6	249	250.5	251.9
	5	1½	383	385	387	389.3	391.5
	7	2	776	780	784	788.5	793
	8	2½	1278	1284	1292	1298	1306
	10	3	2340	2350	2365	2378	2390
	12	3½	3498	3515	3530	3550	3570
	14	4	4939	4966	4994	5023	5051
	15	4½	6721	6757	6796	6835	6873
	17	5	9144	9193	9246	9299	9352
	20	6	15005	15085	15172	15259	15345
	24	7	22228	22346	22476	22603	22731
	27	8	31245	31410	31591	31772	31953
	30	9	42215.3	42439.2	42683.5	42927.7	43172
	34	10	57015.7	57318.1	57648	57977.8	58307.7
	40	12	91509.1	91994.5	92523.9	93053.4	93582
	47	14	118514	119142.5	119828.2	120513.9	121199.7
	53	16	170858.8	171765	172753.6	173742.2	174730.8
	60	18	235124.1	236371.2	237731.6	239092	240452.4
	67	20	312129.5	313785	315591	317396.9	319202.9
	81	24	507320.4	510011.1	512946.4	515881.7	518817

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0455 to .0475 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0455	.0460	.0465	.0470	.0475
75′	2	⅜	12.7	12.8	12.83	12.95	13.0
100′	2	½	24.92	25.1	25.2	25.4	25.5
125′	3	¾	57.2	57.5	57.8	58.13	58.43
150′	4	1	115.15	115.8	116.4	117.15	117.65
	5	1¼	253.3	254.6	256	257.4	258.8
	5	1½	393.7	396	398	400	402.3
	7	2	798	802	806	811	815
	8	2½	1315	1322	1328	1335	1343
	10	3	2408	2420	2430	2442	2457
	12	3½	3595	3615	3630	3650	3670
	14	4	5080	5106	5135	5163	5189
	15	4½	6912	6948	6987	7026	7061
	17	5	9405	9453	9506	9559	9607
	20	6	15422	15502	15589	15675	15755
	24	7	22860	22978	23107	23235	23353
	27	8	32134	32299	32480	32661	32927
	30	9	43416.2	43640.1	43884.4	44128.6	44352.5
	34	10	58637.6	58940	59269.9	59599.8	59902.2
	40	12	94112.3	94597.7	95127.1	95656.6	96142
	47	14	121885.4	122513.9	123199.7	123885.4	124513.9
	53	16	175719.3	176625.5	177614.1	178602.7	179508.9
	60	18	240679.2	243059.9	244420.3	245780.7	247027.7
	67	20	321008.9	322664.9	324470.1	326276.5	327931.8
	81	24	521752.3	524443	527378.4	530313.7	533004.4

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0480 to .0500 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0480	.0485	.0490	.0495	.0500
75′	2	⅜	13.07	13.15	13.2	13.25	13.3
100′	2	½	25.6	25.75	25.85	26.0	26.15
125′	3	¾	58.75	59.07	59.33	59.64	59.95
150′	4	1	118.3	118.9	119.5	120.15	120.75
	5	1¼	260.3	261.4	262.7	264.2	265.4
	5	1½	404.5	406.5	408.5	410.8	412.7
	7	2	820	824	828	832	836
	8	2½	1350	1355	1364	1370	1378
	10	3	2472	2480	2496	2508	2522
	12	3½	3690	3710	3730	3750	3770
	14	4	5218	5244	5270	5297	5325
	15	4½	7100	7136	7172	7207	7246
	17	5	9660	9709	9757	9806	9859
	20	6	15852	15931	16011	16091	16177
	24	7	23482	23600	23717	23835	23964
	27	8	33007	33173	33339	33504	33685
	30	9	44596.8	44820.7	45044.6	45268.5	45512.7
	34	10	60232.1	60534.5	60836.9	61139.3	61469.2
	40	12	96671.4	97156.8	97642.1	98127.4	98656.9
	47	14	125199.6	125828.2	126456.8	127085.4	127771.1
	53	16	180497.5	181403.7	182309.9	183216	184204.6
	60	18	248388.1	249635.2	250882.2	252129.3	253489.7
	67	20	329737.1	331393.1	333048.4	334704	336510.1
	81	24	535939.7	538630.4	541321.1	544011.8	546947.1

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0510 to .0550 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0510	.0520	.0530	.0540	.0550
75′	2	⅜	13.47	13.61	13.75	13.87	14.00
100′	2	½	26.4	26.7	26.9	27.2	27.4
125′	3	¾	60.55	61.15	61.7	62.28	62.87
150′	4	1	121.9	123.15	124.3	125.4	126.65
	5	1¼	268.2	270.7	273.3	275.8	278.5
	5	1½	417	421	425	429	433
	7	2	844	853	861	869	877
	8	2½	1392	1405	1418	1432	1445
	10	3	2546	2570	2594	2620	2643
	12	3½	3805	3840	3875	3915	3950
	14	4	5378	5430	5482	5532	5585
	15	4½	7317	7389	7460	7528	7599
	17	5	9956	10052	10149	10242	10339
	20	6	16337	16496	16655	16806	16966
	24	7	24200	24436	24671	24896	25132
	27	8	34017	34348	34680	34996	35327
	30	9	45960.5	46408.3	46856.1	47283.6	47731.4
	34	10	62074	62678.8	63283.5	63860.9	64465.6
	40	12	99627.6	100598.3	101569	102495.5	103466.2
	47	14	129028.2	130285.4	131542.5	132742.5	133999.6
	53	16	186017	187829.4	189641.8	191371.8	193184.2
	60	18	255983.8	258477.9	260971.9	263352.7	265846.7
	67	20	339821	343131.7	346442.7	349603.2	352914.1
	81	24	552328.5	557710	563091.4	568228.2	573600.6

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0560 to .0600 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0560	.0570	.0580	.0590	.0600
75′	2	⅜	14.13	14.25	14.36	14.5	14.62
100′	2	½	27.65	27.95	28.18	28.4	28.65
125′	3	¾	63.43	64.0	64.58	65.13	65.68
150′	4	1	127.8	128.8	130	131.2	132.3
	5	1¼	281	283.5	286	288.5	290.8
	5	1½	436.5	440.5	444.6	448.5	452
	7	2	885	893	900	908	916
	8	2½	1458	1470	1483	1496	1508
	10	3	2668	2690	2713	2740	2763
	12	3½	3985	4020	4060	4090	4130
	14	4	5635	5685	5735	5785	5832
	15	4½	7667	7735	7803	7871	7936
	17	5	10432	10524	10617	10709	10798
	20	6	17118	17269	17422	17574	17718
	24	7	25357	25582	25807	26032	26247
	27	8	35644	35960	36276	36593	36894
	30	9	48158.8	48586.3	49013.7	49441.2	49848.3
	34	10	65043	65620.3	66197.6	66774.9	67324.7
	40	12	104392.8	105319.3	106245.9	107172.5	108054.9
	47	14	135199.6	136399.6	137599.6	138799.6	139942.5
	53	16	194914.2	196644.2	198374.2	200104.2	201751.8
	60	18	268227.5	270608.2	272988.9	275369.6	277637
	67	20	356074.5	359235.4	362395	365555.8	368565.7
	81	24	578746.4	583883.2	589020	594156.8	599049

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0610 to .0650 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0610	.0620	.0630	.0640	.0650
100′	2	⅜	14.73	14.85	14.98	15.1	15.21
125′	2	½	28.9	29.15	29.4	29.6	29.8
150′	3	¾	66.2	66.8	67.3	67.85	68.35
	4	1	133.35	134.5	135.6	136.65	137.65
	5	1¼	293.2	295.7	298	300.5	302.7
	5	1½	455.8	459.8	463.3	467	470.5
	7	2	923	931	939	946	953
	8	2½	1520	1539	1545	1558	1570
	10	3	2783	2810	2830	2853	2872
	12	3½	4160	4195	4230	4265	4295
	14	4	5880	5930	5978	6025	6071
	15	4½	8001	8069	8134	8199	8260
	17	5	10886	10979	11067	11155	11239
	20	6	17863	18015	18160	18304	18442
	24	7	26461	26686	26900	27115	27318
	27	8	37195	37512	37813	38114	38401
	30	9	50255.3	50682.8	51089.9	51497	51883.7
	34	10	67874.5	68451.8	69001.6	69551.4	70073.7
	40	12	108937.4	109863.9	110746.4	111628.8	112467.1
	47	14	141085.3	142285.3	143428.2	144571	145656.7
	53	16	203399.5	205129.5	206777.1	208424.7	209990
	60	18	279904.3	282285	284552.4	286819.7	288973.7
	67	20	371575.2	374736.1	377746	380756.1	383615.9
	81	24	603941.2	609078	613970.2	618862.4	623509.9

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0660 to .0700 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Deop Per Lineal Foot				
			.0660	.0670	.0680	.0690	.0700
100′	2	⅜	15.33	15.45	15.55	15.67	15.77
125′	2	½	30.05	30.3	30.45	30.7	30.95
150′	3	¾	68.9	69.4	69.9	70.4	70.92
	4	1	138.75	139.8	140.8	141.8	142.85
	5	1¼	305	307.4	309.6	311.8	314
	5	1½	474.2	478	481.3	485	488.5
	7	2	961	968	974	982	989
	8	2½	1583	1594	1605	1617	1630
	10	3	2896	2918	2940	2961	2985
	12	3½	4330	4360	4390	4425	4460
	14	4	6118	6164	6209	6254	6299
	15	4½	8325	8387	8448	8510	8571
	17	5	11327	11411	11495	11578	11662
	20	6	18587	18724	18862	19000	19137
	24	7	27533	27736	27940	28144	28347
	27	8	38702	38988	39274	39561	39847
	30	9	52290.8	52677.5	53064.3	53451	53837.7
	34	10	70623.6	71145.9	71668.2	72190.5	72712.9
	40	12	113349.6	114187.9	115026.2	115864.5	116702.8
	47	14	146799.6	147885.3	148971	150056.7	151142.4
	53	16	211637.6	213202.9	214768.1	216333.3	217898.6
	60	18	291241.1	293395	295549	297703	299857
	67	20	386625.3	389484.6	392344.2	395203.6	398063.3
	81	24	628402.1	633049.7	637697.3	642344.9	646992.5

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0710 to .0750 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0710	.0720	.0730	.0740	.0750
100′	2	⅜	15.9	16.0	16.12	16.22	16.32
125′	2	½	31.2	31.4	31.6	31.8	32.0
150′	3	¾	71.45	71.95	72.42	72.92	73.42
	4	1	143.9	144.85	145.85	146.9	147.9
	5	1¼	316.3	318.6	320.8	323.0	325.0
	5	1½	492.0	495.3	498.7	502.0	505.5
	7	2	996.0	1003	1010	1017	1024
	8	2½	1642	1654	1664	1675	1686
	10	3	3005	3026	3046	3068	3088
	12	3½	4490	4520	4550	4585	4615
	14	4	6345	6390	6433	6478	6521
	15	4½	8633	8695	8753	8814	8873
	17	5	11746	11830	11909	11993	12072
	20	6	19274	19411	19542	19679	19809
	24	7	28551	28755	28948	29151	29344
	27	8	40133	40419	40690	40977	41248
	30	9	54224.5	54611.2	54977.6	55364.3	55730.7
	34	10	73235.2	73757.5	74252.3	74774.7	75269.5
	40	12	117541.2	118379.5	119173.7	120012.0	120806.2
	47	14	152228.2	153313.9	154342.4	155428.1	156456.7
	53	16	219463.8	221029.1	222511.9	224077.2	225560.1
	60	18	302011.0	304165.0	306205.6	308359.6	310400.2
	67	20	400922.2	403782.2	406490.6	409350.0	412059.2
	81	24	651640.1	656287.6	660690.6	665338.2	669741.2

Pounds of Steam Carried in Pipe *for* Drops per Lineal Foot of .0760 to .0800 Lb.

Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0760	.0770	.0780	.0790	.0800
100′	2	⅜	16.42	16.55	16.65	16.77	16.87
125′	2	½	32.2	32.45	32.7	32.9	33.1
150′	3	¾	73.9	74.4	74.87	75.35	75.85
	4	1	148.8	149.85	150.8	151.8	152.8
	5	1¼	327.2	329.5	331.6	333.8	335.9
	5	1½	509.0	512.5	515.3	518.8	522.0
	7	2	1030	1038	1044	1050	1057
	8	2½	1697	1709	1720	1730	1742
	10	3	3108	3130	3150	3170	3190
	12	3½	4645	4675	4705	4735	4765
	14	4	6564	6609	6649	6692	6735
	15	4½	8931	8993	9048	9106	9164
	17	5	12151	12235	12310	12390	12469
	20	6	19940	20077	20200	20330	20461
	24	7	29537	29741	29923	30116	30309
	27	8	41519	41805	42061	42333	42604
	30	9	56097.1	56483.8	56829.9	57196.2	57562.6
	34	10	75764.3	76286.6	76754.0	77248.8	77743.7
	40	11	121600.4	122438.7	123188.8	123983.0	124777.2
	47	14	157485.3	158571.0	159542.4	160571.0	161599.6
	53	16	227012.9	228608.2	230008.6	231491.5	232974.4
	60	18	312440.8	314594.8	316522.0	318582.6	320603.2
	67	20	414767.6	417627.8	420186.0	422894.9	425603.9
	81	24	674144.1	678791.7	682950.1	687353.1	691756.0

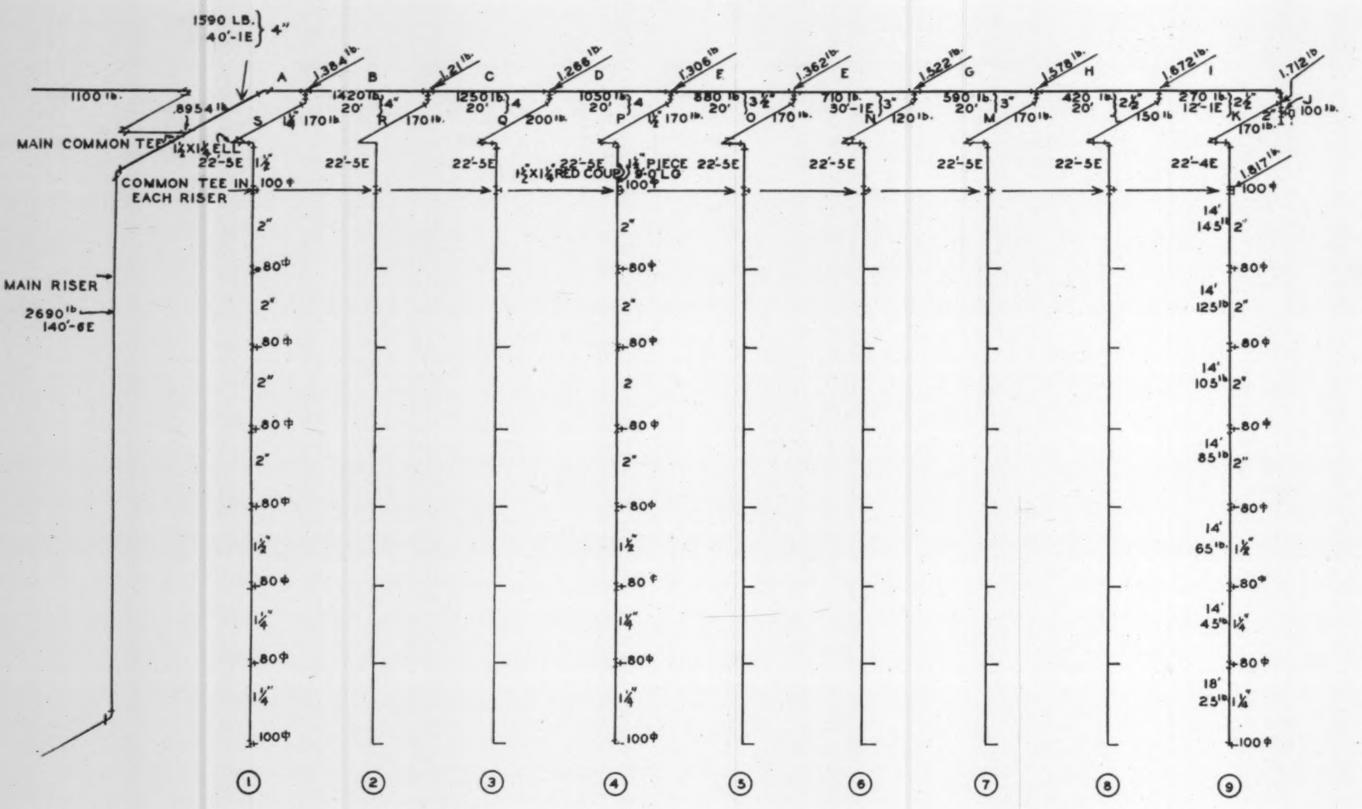

1590 LB.
40'-1E
4"
1100 lb.
.8954 lb
MAIN COMMON TEE
1½X1¼ ELL
22'-5E
COMMON TEE IN EACH RISER
MAIN RISER
2690 lb
140'-6E
A
B
C
D
E
E
G
H
I
J
K
S
R
Q
P
O
N
M
1.384 lb.
1.21 lb.
1.266 lb.
1.306 lb
1.362 lb.
1.522 lb.
1.578 lb.
1.672 lb.
1.712 lb
1.817 lb.
1420 lb. 20' 4"
1250 lb. 20' 4
1050 lb. 20' 4
880 lb 20' 3½"
710 lb. 30'-1E 3"
590 lb. 20' 3"
420 lb. 20' 2½"
270 lb. 12'-1E 2½"
2" 100 lb.
1¼ 170 lb
170 lb.
200 lb.
1½ 170 lb.
170 lb.
120 lb.
170 lb.
150 lb
170 lb.
22'-5E
22'-4E
1½" PIECE 9-0 LG
1½X1¼ RED COUP.
100 ф
80 ф
1½"
1¼"
2"
14'
145 lb
125 lb
105 lb
85 lb
65 lb.
45 lb
18'
25 lb
①
②
③
④
⑤
⑥
⑦
⑧
⑨

Fig. 4—Sizing the Pipe for the Type of Heating System Often Found in Offices, Hotels, Stores, Etc.

amount of 1¼" pipe to install is determined as follows:

170 lb. on 1¼" pipe = .0205 drop per ft.
170 lb. on 1½" pipe = .0085 drop per ft.

.0120 difference in drop per ft.

.0085 × 47 ft. = .3995 drop if connection was all 1½" pipe.

.511 required drop — .3995 = .1115 lb. drop to be added in connection by use of 1¼" pipe.

.1115 ÷ .0120 = 9 equivalent ft. of 1¼" pipe required, the balance of run to be 1½" pipe.

By reference to the diagram for Fig. 4, it will be seen that this throttle in a 1½ in. connection has been accomplished by insertion of a 9-ft. piece of 1¼ in. pipe. The reduction or throttle should always be placed with a view to accuracy of installation rather than otherwise.

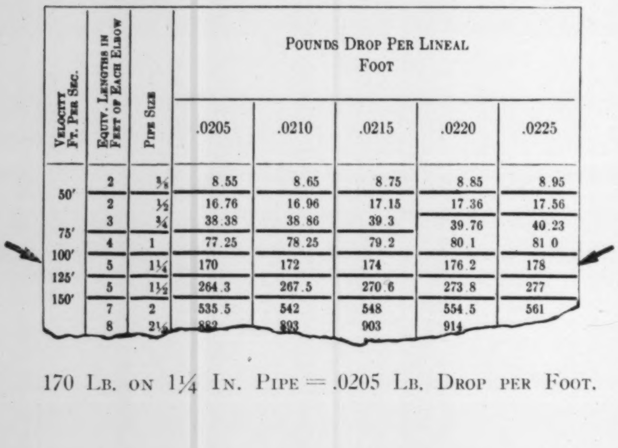
In the diagram has also been shown the results of calculations for connection *S* to riser unit (1), but the details of calculations are omitted.

All that remains of the Third Step is to calculate the sizes of connections to each riser in a similar manner.

Fourth step: The final step in the completion of the calculations for Example No. 2 (Fig. 4) is to figure the other branch main (and series of risers not shown) which is shown taken off at "main common tee," using the same base drop as in the section already figured. For this main as an individual section the actual drop in pressure required from main common tee to first or common tee of each riser would be:

Base drop	1.817
Drop at main common tee	.895
Required drop	.922

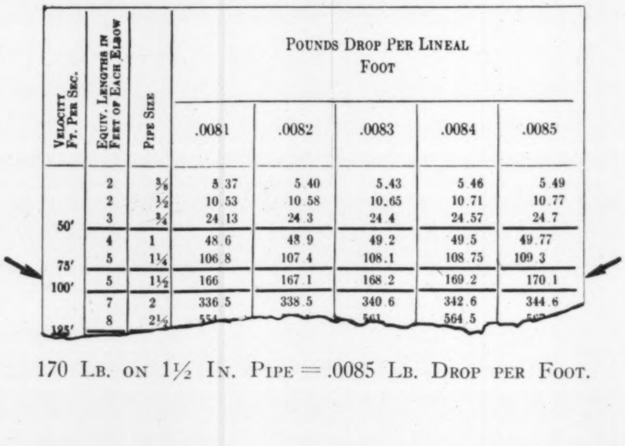
Pounds of Steam Carried in Pipe
for
Drops per Lineal Foot of .0205 to .0225 Lb.



Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0205	.0210	.0215	.0220	.0225
	2	⅜	8.55	8.65	8.75	8.85	8.95
50'	2	½	16.76	16.96	17.15	17.36	17.56
	3	¾	38.38	38.86	39.3	39.76	40.23
75'	4	1	77.25	78.25	79.2	80.1	81.0
100'	5	1¼	170	172	174	176.2	178
125'	5	1½	264.3	267.5	270.6	273.8	277
150'	7	2	535.5	542	548	554.5	561
	8	2½	882	893	903	914	

170 Lb. on 1¼ In. Pipe = .0205 Lb. Drop per Foot.

Pounds of Steam Carried in Pipe
for
Drops per Lineal Foot of .0081 to .0085 Lb.



Velocity Ft. Per Sec.	Equiv. Lengths in Feet of Each Elbow	Pipe Size	Pounds Drop Per Lineal Foot				
			.0081	.0082	.0083	.0084	.0085
	2	⅜	5.37	5.40	5.43	5.46	5.49
	2	½	10.53	10.58	10.65	10.71	10.77
50'	3	¾	24.13	24.3	24.4	24.57	24.7
	4	1	48.6	48.9	49.2	49.5	49.77
75'	5	1¼	106.8	107.4	108.1	108.75	109.3
100'	5	1½	166	167.1	168.2	169.2	170.1
	7	2	336.5	338.5	340.6	342.6	344.6
125'	8	2½	[illegible]	[illegible]	[illegible]	564.5	[illegible]

170 Lb. on 1½ In. Pipe = .0085 Lb. Drop per Foot.

EXAMPLE NO. 3— AN UPFEED SYSTEM

FOR fear that it may not be clear to the reader that this method of calculation for equalization is also applicable to an up-feed system, as well as a down-feed system, an example identical with riser unit (4) of Example No. 2 (Fig. 4) is shown herewith reversed (see Fig. 5), but with the same drop in pressure required in the connection to it so as to present the same difficulties.

In arriving at the sizes for the risers and radiator connections, the same data are used as in the previous Example No. 2 (*A. S. H. V. E.* 1930 *Guide,* page 358), excepting that sizes in Col. *K,* which are maximum capacities for up-feed risers, are used.

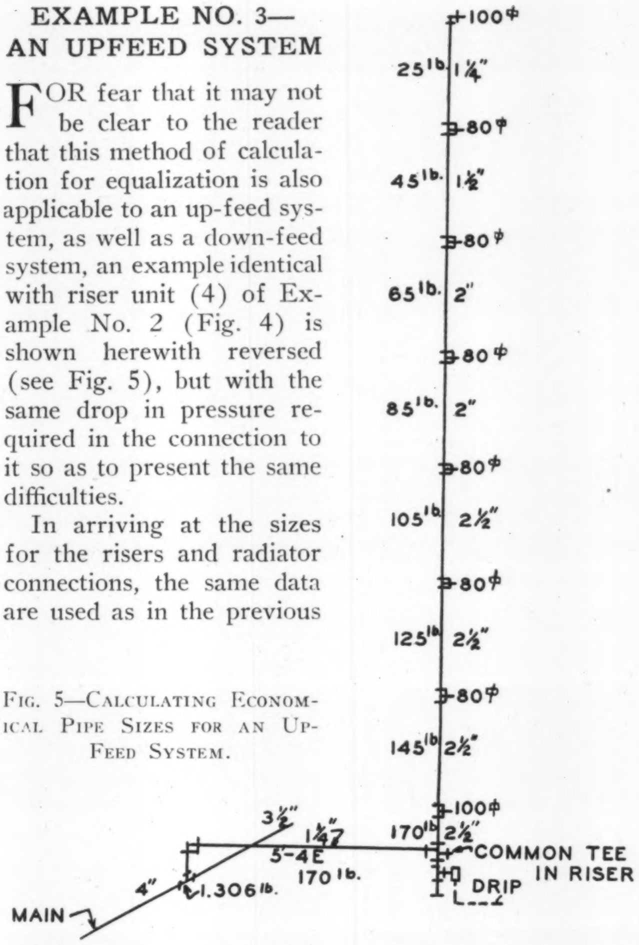


FIG. 5—CALCULATING ECONOMICAL PIPE SIZES FOR AN UP-FEED SYSTEM.

Calculations for connection from main to riser in this case are as follows:

Load 170 lb.
Length 5′—4 ells
.511—Drop required through connection (same as in preceding examples)
On basis—all 1¼″: 5 + 20 = 25′ equiv. length × .0205 = .512 = all 1¼″

It may seem absurd to supply a 2½ in. riser with a 1¼ in. connection, but it is entirely practical as long as every unit is equalized to a practical degree.

M. F. May occupies an office overlooking Chicago's double-decked street, Wacker Drive, and the Chicago River. Recently, he told the following story:

"It was a seventeen-story building. The tenants complained about noise in the radiators. Repeated efforts were made to discover the cause but in spite of all that could be done, the noise continued and, likewise, tenants continued to complain. A contractor made an inspection of the job with no results.

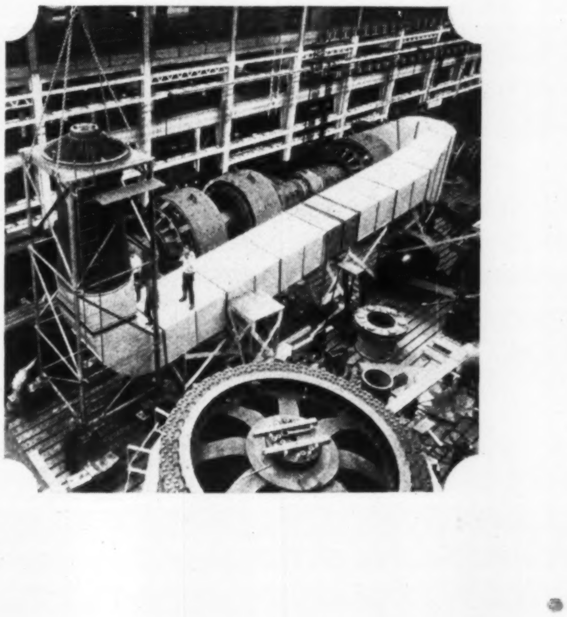
"One morning we had a meeting of the heating contractor, the building engineer, the service man and the radiator man. We found that there was no argument about the noise, as it was certainly present. When the radiator valve was shut off, the noise disappeared but as soon as the valve was opened, there was a steady hissing in the radiator. After checking over several things we finally removed the orifice plates which the manufacturer of the heating system had furnished with his valves so that the system could be properly balanced. As soon as this orifice plate was removed, the noise stopped, with the result that the manufacturer advised the engineer that he would go over the job and change the orifice plates to larger ones, which would eliminate this noise.

"The above condition was caused by the use of orifice plates too small to be used with copper radiation, which does not have the weight and steam chamber of cast iron radiation. Evidently in this type of radiation the noise was amplified very noticeably, whereas if the same orifice plates were used with cast iron they would be satisfactory.

"Since that time the same condition has arisen on other jobs and this has always been the remedy."

Perhaps this will help you solve a similar problem; and the method you used on some other "trouble job" will be of help to your fellow-engineers. Write us, telling us about it, so that it may be published for the benefit of others.

Fan for Blooming Mill Motors



These 5,000 hp. double-unit reversing motors, for twin drive on a blooming mill, will, when installed, deliver 10,000 hp., more than has ever before been applied to a single pair of rolls. The picture at the left shows a shop test setup.

To provide proper ventilation to care for the amount of heat generated by these motors under load the fan shown in the foreground of the photograph is being used to force air through the duct running alongside. It is a vertical propeller type, powered by a 75 hp. motor running at 1450 rpm., supplying 70,000 cu. ft. of air per minute.

The final installation of these motors will follow the marine procedure. Air from outside will be forced into a closed basement beneath the machine and will make its exit through ports into the motors themselves. No ducts or piping will be necessary.

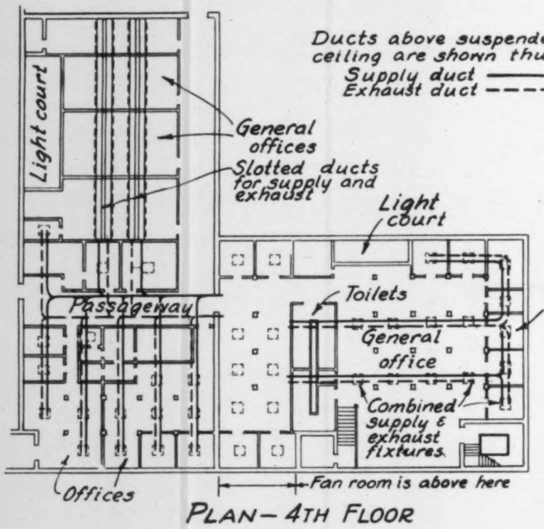
Cooling Libby, McNeill and Libby's General Offices

By Samuel R. Lewis*

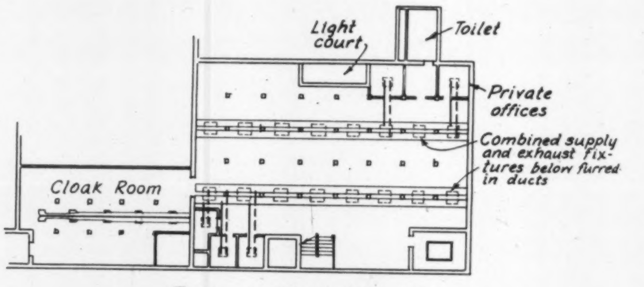
CHICAGO offices of the Libby, McNeill and Libby Company are located on the second, third and fourth stories of an old building. They are not confined to four walls, but occupy different areas on each story which overlap only partially, with reference to the other stories.

A fundamental proposition was that the plant should be designed as a cooling system, heating being entirely separate, and that the entering air for ventilation purposes always should be at least a trifle cooler than the air already in the room. The air cooling equipment would be *above* the spaces to be served, so that it would be possible, to a degree, to *assist* Nature in allowing the heavy cool air to roll and fall down into the rooms.

FIGS. 1 AND 2—THE ROOMS GENERALLY ARE WELL-LIGHTED WITH THE MAJOR EXPOSURE TO THE SOUTH. THE WINDOWS ARE DOUBLE, BUT THERE IS NO BUILDING-INSULATION. THE DUCTS ARE ALL FURRED-IN ABOVE THE SUSPENDED CEILINGS. AN EXHAUST DUCT RUNS BESIDE EVERY SUPPLY DUCT. HEATING IS BY DIRECT RADIATORS, CONTROLLED BY THERMOSTATS



PLAN—4TH FLOOR



PLAN - 3RD FLOOR

Another fundamental proposition was that an air washer using pumped recirculated water should be used to absorb the heat and moisture from the air, this being an efficient heat-transfer agent, but that the air-washer should not be expected to do any air cleaning.

An automatic air filter, using a constantly-renewed film of heavy oil over progressive filter bodies would be depended upon for removal of dust and soot.

It would have been desirable to place the exhaust fan *below* the offices, carrying on consistently the idea of helping Nature in the flow of the relatively heavy cool air, but this location was not available, so the exhaust fan was perforce placed alongside the supply fan above the offices.

Exhaust Fan Required Increased Speed

It is of interest to observe here that the supply fan delivered the calculated volume of air at the calculated speed, but that the exhaust fan, due apparently to consistently handling cooler air than that normally handled by conventionally-located exhaust fans, fell down lamentably as to its original capacity as far as could be judged by anemometer and pitot tube observations, and eventually required a larger motor and a much-increased speed.

In this connection it is fair to state that the exhaust fan outlet air speed was too high, and the space too constricted for any conclusive anemometer reading of the average velocity.

The outlets in the rooms, on the other hand, were so many and so large and so had such a low velocity that anemometer readings again were untrue and inconclusive.

Smoke tests and sense-impression tests indicated excellent ventilation and a lively air movement at many grilles which had too low an air current to turn the anemometer wheel.

Anemometer Tests difficult

All Chicago ventilating systems must be approved, after the air flow has been measured and found satisfactory by anemometer, by a health department inspector. The inspector in the case of this plant had a difficult task, as the peculiar low-velocity outlets and inlets, many in number in all of the larger rooms, and the narrow slots in some of the other rooms seemed to have been made (without any intention of so doing), to render his task difficult.

As now operated it seems apparent that the exhaust fan exerts some excess pressure at the outlets nearest to it, as on the fourth story, especially in cool weather, there is a lively movement of air from the third story

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* Consulting engineer, Chicago.

The air conditioning of offices has been found to pay dividends in increased efficiency of the occupants—soon, every office will be cooled and dehumidified in summer, warmed and humidified in winter.

up to the fourth story through an open interior stairway.

It was observed that when cooling the offices in warm weather this updraft becomes much less noticeable, another confirmation of our experience as to the difficulty of lifting cooled air up to an exhaust fan in warm weather.

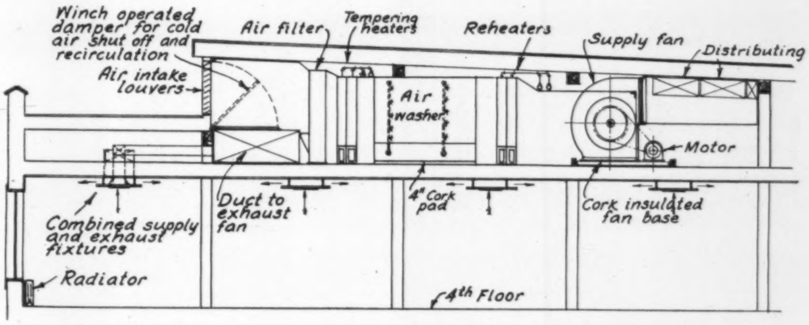


Fig. 3 — The Air Conditioning Equipment Is Above the Rooms Served by It. The Air May Be Recirculated When Refrigeration Is in Effect. The Refrigerating Plant and Water-Cooler Are in the Basement, Cold Water Being Pumped Therefrom to the Air Washer. Automatic Air Filters Are Out-Board from the Washer

Construction of Offices

The ceilings of the offices are comparatively low, and were broken up by beams and columns galore, the source of most trouble of earlier cool air diffusing schemes for similar large areas. The installation of comparatively smooth suspended ceilings to cover the beams was made to eliminate such trouble as much as possible.

There are many private offices, and many of these are surrounded by glass partitions, so that flue-locations were very hard to find.

Since the structure is old and of non-fireproof construction, it was not possible to extend duct-risers alongside or around or inside the columns, as has occasionally been done elsewhere under similar conditions.

Introducing the Air

Taking into consideration the fundamental determination to use a supply of air always cooler than that already at the zone of the room-inlet, a combined inlet and outlet fixture, which should discharge the cool air supply at rather high speed horizontally overhead, and which would remove the spent air at rather low velocity through a downward-looking grille, also overhead, was devised.

If such a scheme would work, and if one of these combination fixtures could be placed in each bay, much of the difficulty from drafts due to beams and columns could be avoided and the walls would not be cluttered up with flues.

Such a combination supply and exhaust fixture for each private office would settle the matter for that office, and would leave the glass partitions clear. (See Fig. 5.)

Private Offices Usually Too Cool

Experience has shown that private offices around the edges of general offices, when given an air supply at the same temperature as the general offices, when cooling, will be too cold. It is difficult and costly to arrange air cooling systems with separate ducts to each room, each having a thermostatically-operated double mixing damper, as would be the most desirable manner in which to solve this problem.

Fortunately at Libby, McNeill and Libby's, steam is available the year 'round, and any too-cool private offices at any season or temperature can be taken care of by

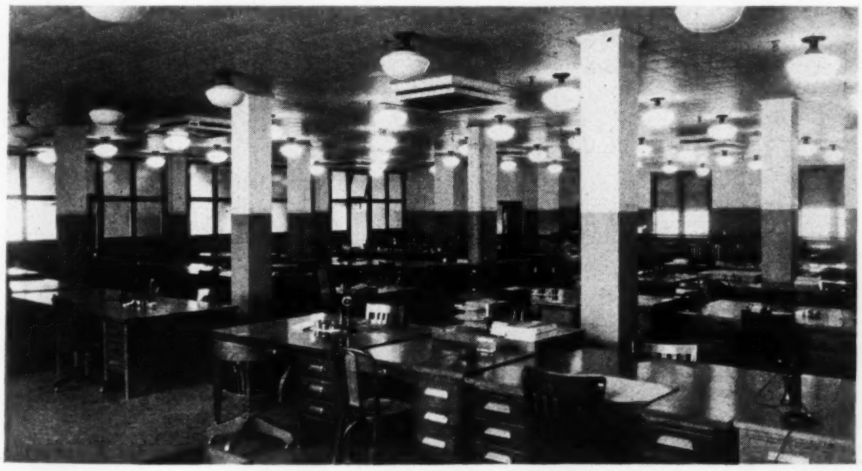


Fig. 4—Fourth Story General Office. The Supply and Exhaust Combination-Diffusers Are Shown in the Center of Each Bay. The Ceiling and the Diffuser-Bottoms Are Treated for Sound-Absorption. Private Offices, Each Having Similar Combination-Diffusers, Are Along the Left

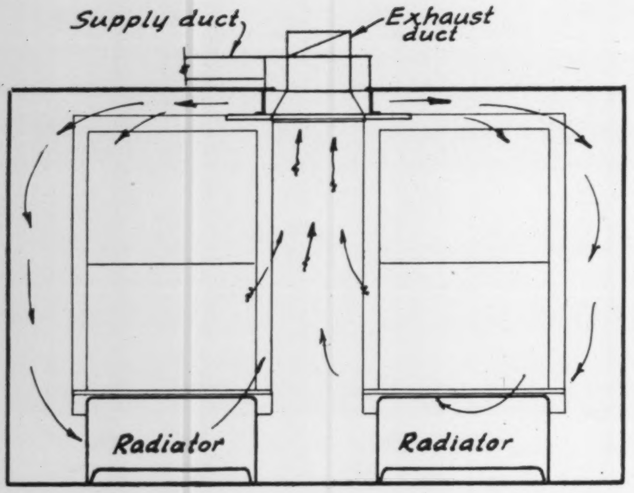


Fig. 5—Section Through Combination-Diffuser and One Bay Served by It. The Supply Is Delivered Horizontally Above the Wide Bottom Plate. The Exhaust Is Withdrawn Vertically Through the Center of the Bottom Plate. Supply and Exhaust Ducts, Both Serving the Combination-Diffuser, Are Above the Suspended Ceiling

allowing the thermostat to open the radiator valve in the room.

For certain rather isolated offices slot air-introduction and removal was used, with the supply and exhaust ducts alternated. It was found that the air rather hated to leave the slots in an orderly manner, but all wished to leave at the remote end of each room-slot. This was corrected, but the general impression from the slot-experience was that it would have been better to use the combination-diffusers in this case.

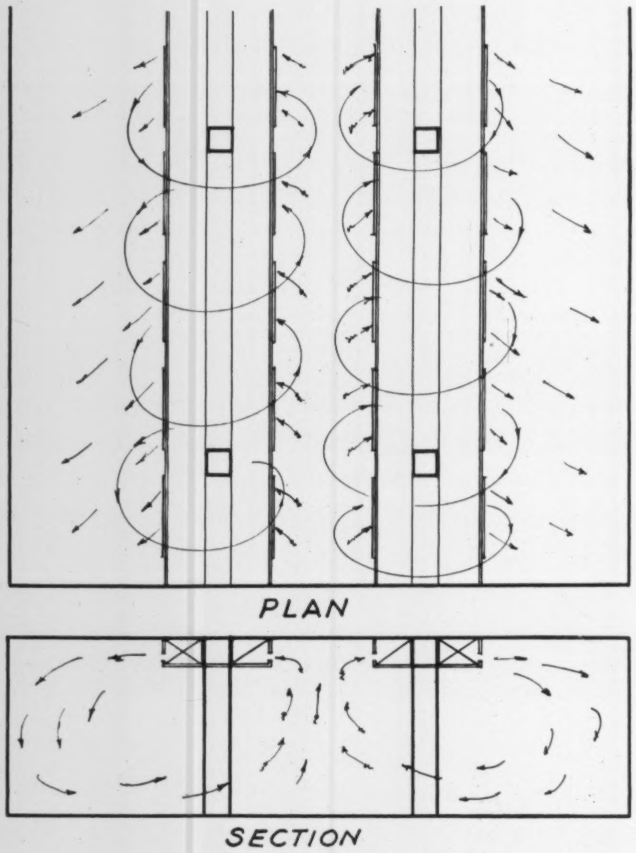


Fig. 6—The Slot-Distribution as We Intended It to Be, the Cool Air in a Thin Sheet Falling Toward the Floor, Picking Up Heat, and Rising to Be Exhausted at the Ceiling Through a Slot Opposite the Inlet Slot

Our experience with 1½ in. high continuous slots for air inlet and air outlet has, in general, been very favorable.

Broadcasting Rooms

The broadcasting rooms at the Libby plant are entirely interior spaces, with walls, ceilings and floors of sound-proof construction. They use side-looking supply grilles and down-looking vents. The ducts to these rooms have lined and faced labyrinths which effectively have stopped any objectionable sound-transfer.

In some of the private offices, for fear of possible draft down the walls due to reaction from the horizontally-looking supply grilles, the combination-diffusers were reversed, so that the air supply came down vertically through the normal vent grilles, while the exhaust was taken through the normal horizontally-looking supply grilles.

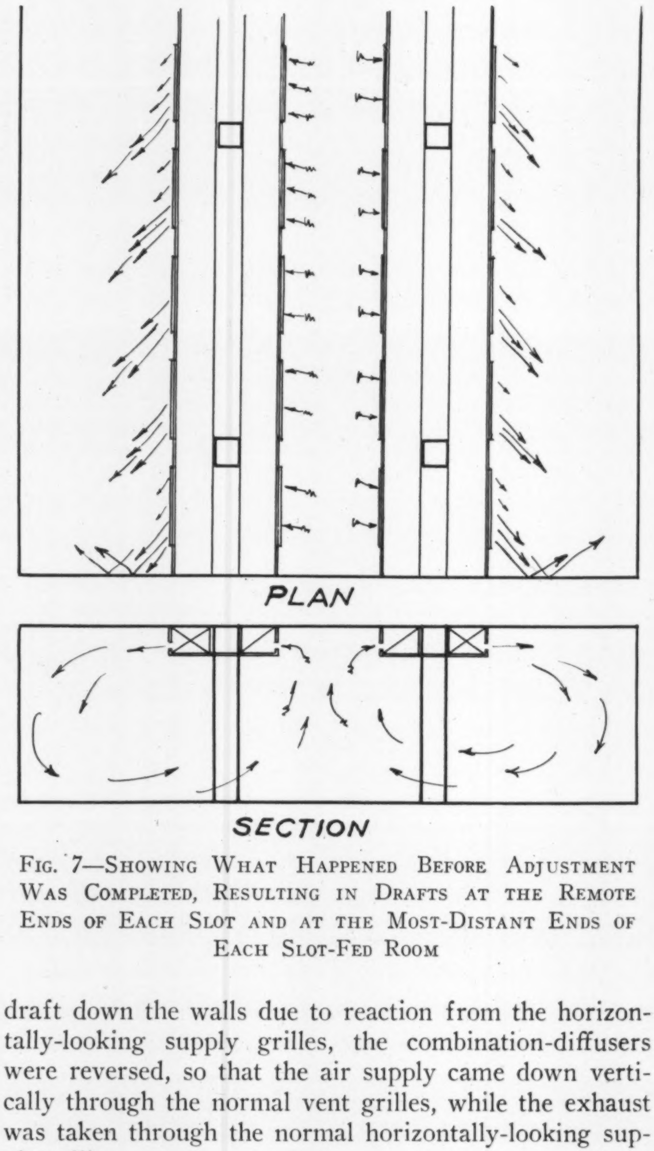


Fig. 7—Showing What Happened Before Adjustment Was Completed, Resulting in Drafts at the Remote Ends of Each Slot and at the Most-Distant Ends of Each Slot-Fed Room

This was a mistake, as the incoming air caused draft-complaint. In a few cases the diffusers could not conveniently be changed back, and these were made very satisfactory by the installation of additional interior deflectors which deliver the supply air in a cone-shaped spray.

Humidifying in Winter

The air washer is used for humidifying as required in winter by adding warm water to the circuit. In summer the cooling is achieved by pumping the water through the tubes of a closed cooler placed in the basement. The cooler is served by a 125-ton ammonia compressor.

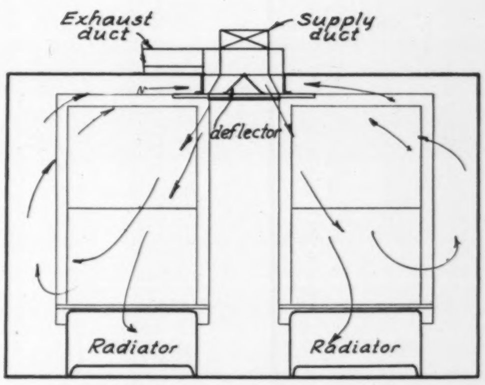


Fig. 8—Section Through Reversed Combination-Diffuser and One Bay Served by It. The Downward-Going Cool Air Was Unbearable and Additional Glass Diversion-Plates or Inserted Cone-Shaped Deflectors Were Required. The Resulting Air Currents Were Satisfactory

The circulating pump for the washer, which also generates pressure for the spray, is placed in the basement near the cooler.

Provisions are made so that during certain short periods of the year, while the city water is cooler than the desired air delivery temperature, this water can be used in the washer, warming the water a few degrees. The waste water then passes on to be used for certain process work where the added temperature and possible, though doubtful, other contamination, can do no harm.

This plant handles about 42,000 cu. ft. of air per minute, and involves an investment of about $50,000.00.

"Open for Discussion"

A department in which we follow the custom of technical societies in allotting space in their programs for discussion of the papers presented

Windowless Buildings

Cost of Heating and Cooling Windowless Buildings, by *Harold L. Alt*

SUGGESTIONS have been advanced from time to time that the tall building of the future may develop into a windowless structure and it is not entirely beyond the realms of possibility that such, indeed, may prove to be the case. One of the basic ideas back of this suggestion is that air conditioning will be more easily and economically effected in such a building than in one provided with the usual number of windows. That there are certain advantages to be obtained by this means can not be denied, but there are also some disadvantages.

As with all other items connected with new building construction, cost occupies a most important position in regard to the matter of supplying or eliminating windows. In the consideration of any particular item in the design of a building the first cost plays a most prominent part with the operating cost or upkeep coming close behind. If the first cost is high, it is always difficult to secure the adoption of any particular item over one of lesser cost which provides practically the same results; and, if the first cost is so high that the interest and upkeep charge amount to more than the financial advantage obtained by using the more expensive item, it is practically impossible to win over either the architect or the owner to any plan embodying a deviation from standard methods in building construction.

It is the purpose of this discussion to assume a set of conditions as nearly typical as possible and to demonstrate just what effect a windowless building would have on the mechanical equipment provided for such a building when compared to a similar structure of ordinary construction. This involves not only initial installation cost, but operating costs as well, and it will be seen that, whereas a saving may be made on the initial outlay, it does not follow necessarily that a saving will be made in the cost of operation.

Objections

Among some of the objections and entirely aside from the matter of cost (which will be considered later in this article) might be mentioned the loss of architectural effect by the elimination of windows and the psychological effect on the occupants of a windowless building. True, many buildings already contain "inside" rooms, totally without windows and outside light, but such rooms are nearly always connected, either directly or indirectly, with the outside windows of the building and on the same floor either through corridors or by means of doors opening into adjoining spaces where outside windows exist. It is questionable, indeed, if the elimination of all windows is not going to duplicate the conditions found in safe deposit vaults located below the street level where some find that any system of ventilation still leaves something to be desired. Certainly the light and airy conditions obtained on some of the upper floors can not be said to be duplicated in the deposit vault even though the conditions there may be perfectly healthful as far as the support of life may be concerned.

The difficulty of ventilating basements in department stores is well recognized, and what are these basements but an example of windowless construction? It must not be forgotten that the best and tightest window—even when closed—furnishes a means for a certain amount of natural leakage into the building and—while an equal amount of conditioned air may be delivered into the same space by an up-to-date ventilating system—it is indeed questionable if the psychological effect on the inmates is not going to approximate in some degree the sensations produced by entering the vault or basement above cited.

But let it be assumed that these features can and will be overcome by suitable and adequate provisions made in the ventilating and air conditioning system and that the cost item is the only one requiring serious consideration. What then?

Owing to the fact that every building is a separate and individual proposition by itself, it will be necessary to assume some typical case and to work out the comparison on that basis. The results, as long as the type and area of the buildings do not vary too greatly, should be fairly comparable and indicative as to what the various changes and alterations would mean.

For example, let a building 300 ft. long and 200 ft. wide be assumed such as is illustrated in Fig. 1. The total floor area (allowing about 12 in. of wall thickness) will be $(200 - 2) \times (300 - 2)$, or 198 ft. × 298 ft., or 59,004 sq. ft., say 59,000, and the probable number of occupants will be about in the ratio of one occupant to every 100 sq. ft. of floor area. So the occupants will approximate

$$59{,}000/100, \text{ or } 590 \text{ persons.}$$

This figure will vary greatly, of course, in different parts of the building, but it is likely to be very close to the average for the whole building.

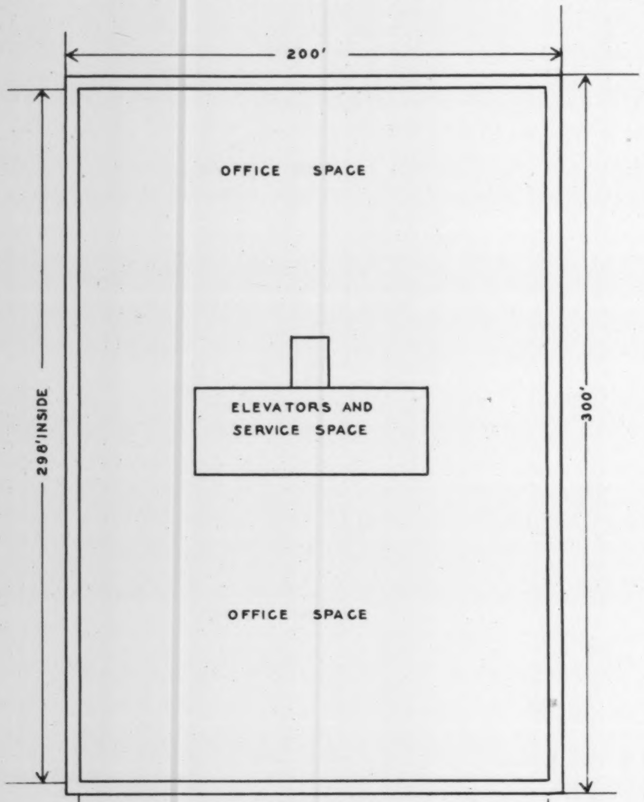


Fig. 1—Typical Floor Plan of an Office Building

Steam Saving

Now the perimeter of the building is 300×2 plus 200×2, or 600 plus 400, or 1,000 lin. ft. and, if the floor heights are about 15 ft., the gross wall area is 1,000 ft. × 15 ft., or 15,000 sq. ft.

Ordinarily, about 1/3 of the gross wall area would be glass (except, of course, on the ground floor, where the ratio would be higher) and the heat loss from the glass—with 70 F inside and zero outside—would be

$$1.1 \times (70 - 0) = 77 \text{ Btu per sq. ft. per hr.}$$

while the wall loss may be assumed as in the neighborhood of

$$0.29 \times (70 - 0) \text{ or } 20+ \text{ Btu per sq. ft. per hr.}$$

Now, if windows are omitted and the ordinary wall construction is to be substituted, there is a reduction of heat loss amounting to

$$77 \text{ Btu} - 20 \text{ Btu or } 57 \text{ Btu per sq. ft. per hr.}$$

It has already been settled that the glass area will be about 1/3 of the gross wall area, and this amounts to

$$1/3 \text{ of } 15{,}000 \times 57 \text{ Btu, or } 285{,}000 \text{ Btu,}$$

but, as some of the glass and wall have north, east, and west exposures, an average exposure factor of 10 per cent will be added, making the actual amount of heat saved

$$285{,}000 \text{ Btu} \times 1.10, \text{ or } 313{,}500 \text{ Btu.}$$

This, reduced to equivalent square feet of direct steam radiation (E. D. R.) gives

$$313{,}500/240, \text{ or } 1{,}306 \text{ sq. ft.}$$

In other words, the radiation required for the typical floor under consideration would be 1,306 sq. ft. less without windows than it would be with windows. Average figures for yearly steam consumption per sq. ft. of steam radiation in climates where the lowest outside temperature hits zero run around 500 lb. of steam per annum; on this basis the saving in steam consumption per year per floor would be

$$1{,}306 \times 500 \text{ lb., or } 653{,}000 \text{ lb. of steam}$$

and, if steam costs to produce in the building or by purchase from outside sources, say, $0.70 per M. lb., then the amount saved in operating expense would be

$$\frac{653{,}000}{1000} \times \$0.70 \text{ or } \$457.10$$

This indicates that it would not take many stories in the building to multiply this amount into something that represents a very big saving.

Additional Electric Consumption

Against this saving in steam is the item of using electric light all day long and every day throughout the year for areas which, otherwise, would receive natural light through the windows. To approximate this, the diagram shown in Fig. 2 has been used; reference to this diagram will show that—roughly—all the floor area over 50 ft. distant from the outside wall has been considered as requiring artificial lighting all the time any way and as not entering into the problem. The floor areas lying within 50 ft. of the outside wall, however, have been considered as not requiring artificial illumination during daylight hours. Even in this area there are certain times when artificial lighting would be required, so what must be computed is the amount of additional lighting which would be required in this portion of the floor by the omission of windows and the cutting off of all outside light.

Let the number of additional hours which the lighting would be used first be calculated; if the working period is assumed at 7 hours and the lights are turned on an average of 15 minutes before and after working hours, the total hours per day would be

$$\tfrac{1}{4} + 7 + \tfrac{1}{4} = 7\tfrac{1}{2} \text{ hr.}$$

but, since the use of the lights during short and dark winter days is bound to occur anyway, running from about an hour a day in the winter down to nothing during the summer, it will be necessary to allow for an average use of ½ hr. per day all the year round, and to subtract this time, when the lights would be used with windows in, from the total time the lights would be used with the windows out, to get the difference or additional hours. Thus:

$$7\tfrac{1}{2} \text{ hr.} - \tfrac{1}{2} \text{ hr.} = 7 \text{ hr.}$$

additional, which, with 300 working days per year, gives

$$7 \text{ hr.} \times 300, \text{ or } 2{,}100 \text{ hr. per year.}$$

The floor area in which this additional use results will be seen (by referring to Fig. 2) to consist of two rectangles, each 50 ft. wide by 198 ft. long lying along the short dimension of the building, and two similar rectangles, also 50 ft. wide by 198 ft. long, lying along the longer sides of the building and gives a total floor area affected of

$$198 \text{ ft.} \times 50 \text{ ft.} \times 4 = 39{,}600 \text{ sq. ft.}$$

which, at a watt per sq. ft.—ordinarily allowed for good lighting—and 1,000 watts to a kilowatt requires

$$39{,}600 \times 1 \text{ watt}/1{,}000 = 39.6 \text{ kw.}$$

It already has been found that the lighting in this area will be used for an additional period of 2,100 hr., when windows are omitted, so that the increased electrical consumption is computed as follows:

$$39.6 \text{ kw.} \times 2{,}100 = 83{,}160 \text{ kw-hr.}$$

and the cost of this extra current at 2 cents per kilowatt hour would be

$$83{,}160 \times \$0.02 = \$1{,}663.20$$

Reduction Due to Lesser Radiator Surface

It may be seen that a certain saving is effected by the omission of 1,306 sq. ft of E.D.R. per floor, not only in the first cost but also in the interest and depreciation charge. If the modern steam heating system runs about $2.00 per sq. ft. installed, the reduction of 1,306 sq. ft will result in a lowering of the first cost by the amount of

$$1{,}306 \times \$2.00 = \$2{,}612.00$$

and the reduction in interest and depreciation charge on this lesser amount of surface—allowing 10 per cent per annum—will be

$$\$2{,}612.00 \times 10 \text{ per cent} = \$261.20$$

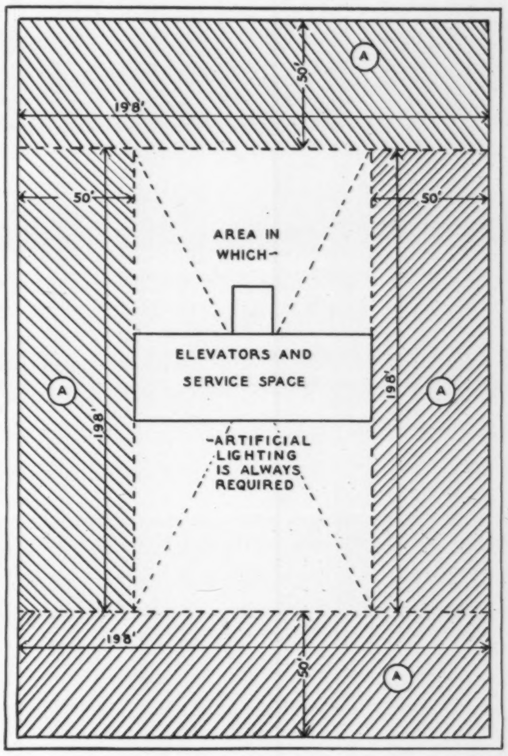

A
50'
198'
50'
50'
AREA IN WHICH—
ELEVATORS AND SERVICE SPACE
A
198'
198'
A
—ARTIFICIAL LIGHTING IS ALWAYS REQUIRED
198'
50'
A

Fig. 2—Plan Showing Spaces in Which Artificial Lighting is Necessary Due to Omission of Windows

Of course, there are other items of credit which are somewhat more difficult to approximate. If the building generates its own steam, for instance, the boilers will be smaller and the space occupied by the boiler room will not be as great as otherwise required, thus making a slightly greater available renting space in the basement. Possibly the operating force may be lowered by a man on each shift owing to the lesser amount of fuel to be handled; certainly a smaller refrigeration plant could be used for summer cooling.

Reductions on the Cooling Plant End

The effect on the cooling plant can be approximated by assuming that the maximum temperature difference between the inside temperature in the building and the outside temperature would not be over 30 F in extreme hot weather, and recalculating the Btu loss on this temperature difference would give

$$30/70 \times 57 \text{ Btu, or } 24+ \text{ Btu per sq. ft. per hr.}$$

and with a total glass area of 5,000 sq. ft., the Btu transmitted in the summer would be

$$24 \text{ Btu} \times 5{,}000 = 120{,}000 \text{ Btu per hr.}$$

One ton of refrigeration capacity is usually taken as 12,000 Btu per hr., so that the capacity of the refrigeration machinery may be reduced 120,000/12,000 = 10 tons per story.

The cost *per ton* on an air conditioning plant recently designed for an office building was about $375.00, so that the saving in dollars in the case under consideration would be

$$\$375.00 \times 10 = \$3{,}750.00$$

which is an initial cost reduction. In addition to this is the smaller amount of electric power required to operate the compressor which—while varying considerably for different operating conditions—will be taken as about 1 1/3 hp. per ton of refrigeration, and the reduction in interest and upkeep amounting to about 10 per cent of the reduction in the cost of the plant; both of these items are operating cost reductions.

For the saving in electric current it will be assumed that the cooling system operates about 3 full months time per year and that the average load is around 50 per cent of the full load. Expressed in days, this time would amount to 3/12 of 300 days, or 75 working days per yr., and it probably would be necessary to run the air conditioning system slightly longer than the actual working hours, say 9 hours per day. This gives the total hours of operation as

9 hr. × 75, or 675 hr.,

and at 50 per cent load the horsepower reduced would be

10 hp × 50 per cent = 5 hp.

which, changed to kilowatts, would be

5 × 0.746, or 3.73 kw.,

so that the total reduction in kilowatt hours will be

3.73 kw. × 675, or 2,518 kw-hr.

With electricity worth two cents per kilowatt hour, this would make the saving in electric current for operating the compressors about.

2,518 × $0.02, or $50.36.

Of course, the saving in interest and depreciation is simply $3,750.00 × 10 per cent, or $375.00.

Leakage

There is no doubt that the elimination of windows will reduce the amount of leakage or "air change" which, otherwise, would occur but as this leakage was already insufficient to meet all the needs of the occupants for ventilation purposes, it cannot be considered as a net reduction owing to the fact that it must be compensated for by increasing the amount of air supplied by the ventilation system accordingly. From this it will be evident that, while a reduction of the amount of surface on the floor should be made to allow for the reduction of leakage when windows are omitted, an increase in the heating surface in the ventilating heater also will have to be considered; for the purpose of this article one has been assumed as practically counterbalancing the other.

When recirculation is employed, there would be a small saving in not having window leakage to consider, even though there would have to be a slightly larger amount of recirculated air. There would be still another small item of power saving on the circulation pump if one were used.

Item of Building Construction

As far as the building construction is concerned, the difference between solid masonry walls and the ordinary windows will have to be computed for each case. It is not difficult to see that where a plain brick wall is considered, it might be cheaper to build a solid wall than to provide windows; but, if a granite or ornamental terra cotta faced wall is substituted, it might easily cost more than the sash which it replaces. For this reason it is impossible to make either a credit or charge item out of the building construction and, for the purposes of this article it will be ignored.

Summary

It now is possible to set down the various amounts which have been calculated and to find out how this proposition of omitting windows lines up from a first cost and operating view point.

These figures indicate that while there would be a net reduction in the initial investment, there would at the same time be a net increase in the operating expense even after the credit for the smaller interest charges and upkeep have been allowed. It is only fair to point out, however, that the increase in operating cost is entirely caused by the extra lighting current consumed in lighting the floor areas which with windows, would receive natural light from the outside. Now, if the building were located in the middle of a block and the two 200 ft. sides were up against another building, the amount charged to extra electric current for lighting would be cut in half but the credits would also be reduced by almost a proportional amount; similarly the difference would be in the neighborhood of only $300.00 per floor, but it could never show an actual saving in operation. For this reason the writer believes the windowless building is not going to develop as fast as it might if the conditions were reversed and a substantial operating reduction could be demonstrated.—*Harold L. Alt.*

COMPARISON OF FIRST COST OR INSTALLATION AMOUNTS

(When Windows Are Omitted)

Item	Increase	Decrease
Reduction in radiator surface	....	$2,612.00
Reduction in air cooling equipment	....	3,750.00
Total	none	$6,362.00

COMPARISON OF OPERATING CHARGES

Item	Increase	Decrease
Lesser steam consumption	....	$ 457.10
Extra lighting current	$1,663.20	....
Lesser compressor current	....	50.36
Reduced overhead on radiators	....	261.20
Reduced overhead on air conditioning equipment	....	375.00
Total	$1,663.20	$1,143.66

Net Rise in operating cost.......... $ 519.54 per floor.

Hay Fever

"Curing Hay Fever with Controlled Weather," *by R. F. Morrison*. Page 412, May, 1930, issue

THE article in the May issue by R. F. Morrison, entitled, "Curing Hay Fever with Controlled Weather," presents some interesting thoughts concerning treatment for this disease, but one of the author's assumptions seems a little over-optimistic.

Experiments in the development of an air filter to remove pollen from atmospheric air have shown that these air-borne irritants are very difficult to catch. And it is particles of this size, the order of twenty microns, and smaller, including other dusts as well as pollen, that offer the greatest health hazard, as Prof. Drinker pointed out in an article in the January 1930 issue of your publication. ("Hygienic Aspects of the Industrial Dust Problem.")

Mr. Morrison, in advocating air conditioning apparatus for hospital treatment of hay fever, says, "Washed air, in a hospital, would eliminate all dangers from irritants in the nature of dusts, fumes, and, to a considerable extent, odors." There is no question that washed air would be a great improvement over unwashed air, but there is considerable doubt as to the ability of a washer to remove the finer particles, which are the most undesirable, from the standpoint of treatment for diseases of the respiratory tract.

The point to be emphasized is that while air washing apparatus is of great benefit in producing the proper humidity, and in removing the larger particles in atmospheric air, it is too much to expect the same apparatus to remove the microscopic particles of pollen, and other dusts. The use of a fabric filter for this purpose has proven of great benefit in such hospitals as Mt. Sinai in Cleveland and the Mayo Clinic at Rochester, Minn., as well as in a large number of offices and private homes.—*E. H. deConingh,* The Dust Recovering & Conveying Co.

Dye Houses

"Unit Heaters Prevent Condensation in Dye Houses," *by John R. Cooper*. Page 767, September, 1930 issue

IF YOU will refer to the elevation plan, Fig. 1, in my article in the September issue, you will note arrows indicate air direction that would seemingly suggest the unit heaters were used for exhaust purposes, rather than supply.

If you will refer to the article in question, you will note that the problem is one of supplying fresh air, properly heated and that the units were hung below the truss line.

In this connection, perhaps, I should have mentioned that the air was delivered horizontally, in order to keep any air movement from disturbing the working force.

Since some comment will, unquestionably, be raised in this particular, I would appreciate your advising anybody interested that the arrows should be reversed, indicating air being taken in at that point, instead of being exhausted and that the air direction of the delivered air is horizontal, below the roof trusses.—*John R. Cooper,* Thermal Units Co..

Group Knowledge

"Pooling of Information." Page 803, September, 1930, issue

THIS editorial calls attention to a characteristic of American engineers that has existed for some years, although it has not always been recognized. The appreciation of the value of group knowledge was the basis for the founding of many of the national societies. A perusal of the objectives of these societies as stated in their charters indicates that they were founded for mutual improvement through the reading of papers, discussions, and the interchange of information and data. This idea is still a part of their programs, the members providing group assistance through the presentation of papers on work done, through committee work, and through the preparation of codes. Our trade journals have also been an important factor in the spreading of group knowledge as they have presented all advances in their various fields as rapidly as these occur.

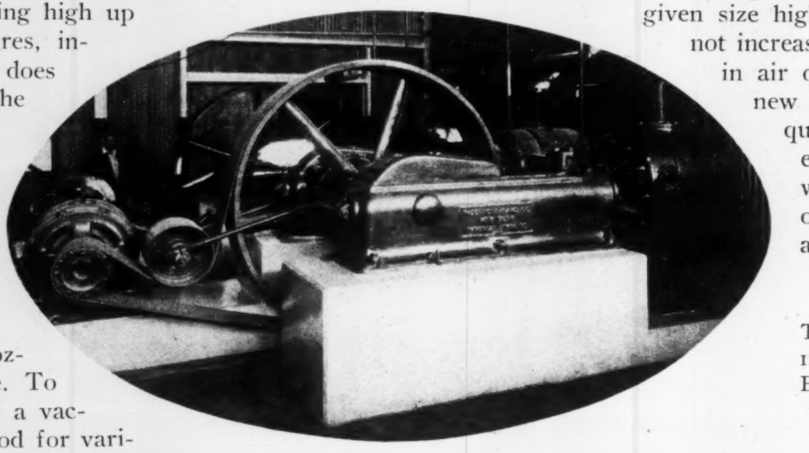
Probably one of the best examples of the value of group knowledge has been in the power plant field. Since the war the progress in American central station development has been phenomenal, and, at the present time, these plants are world leaders. This came about through the rapid adoption of pulverized coal; the use of bleeder heaters; and the application of heat recovery apparatus in new plants. These developments coupled with the use of increased pressures and temperatures have greatly diminished the fuel requirements per kw-hr. of a modern plant. Information on all of these modern developments has been presented from time to time in the reports of the Prime Movers Committee of the *N. E. L. A.* as well as in papers before national societies and in trade journals.

It would seem that similar rapid developments could be brought about through a wider spread of group knowledge in the fields of air conditioning, heating and piping.—*A. G. Christie,* Professor, Mechanical Engineering, The Johns Hopkins University.

How to Make Nozzle Tests of Vacuum Pumps

By S. B. Redfield*

MEASURING the air handling capacity of a vacuum pump is the same kind of a problem as measuring the air delivered by an air compressor. A vacuum pump is merely an air compressor working between pressures lower in the scale than the usual compressor. In each case, the machine takes in its air at a certain intake pressure and discharges it at some higher pressure; the cycle of admission, compression and discharge is the same in both cases; the fact that the vacuum pump admits at a pressure below atmosphere makes no fundamental difference. There are many thousands of "booster" compressors in use throughout the gas fields, taking in gas at 100 or more pounds intake pressure and discharging at considerably higher pressures. These machines are also compressors; but are working high up on the scale of pressures, instead of low down, as does the vacuum pump. The general principles of operation are identical in all cases.



THIS COPPER COMPANY IN ARIZONA USES A BELT-DRIVEN, TWO-CYLINDER VACUUM PUMP

To measure the capacity, or volumetric efficiency of a compressor discharging above atmosphere, an excellent device is the nozzle on the discharge side. To measure the capacity of a vacuum pump a good method for various reasons, is to use a nozzle on the suction side. For all air measurements the nozzle having a low expansion ratio is the best for many reasons; that is, a nozzle of such a size that it will pass all the air to be measured with an absolute pressure on the upstream side not more than twice the absolute pressure on the downstream side. This is close to the *critical* expansion ratio and the method of figuring the quantity of expansion ratios above the critical is different from the method for expansion ratios below the critical. The reason is that for expansion ratios greater than the critical, the actual pressure in the nozzle throat is always that corresponding to the critical ratio, independent of how much lower the down-stream pressure may be. From this is follows that for a given up-stream pressure, only one certain amount of air will flow through an orifice or nozzle, no matter how low the down-stream pressure may be, as long as that down-stream pressure is below the critical pressure; that is, less than half the absolute up-stream pressure.

Contrary to the above, as long as the expansion ratio through the nozzle remains less than about two, a decrease in the down-stream pressure will result in greater flow through the nozzle.

From this it will be seen that for vacuum pump testing on the inlet side, the low ratio nozzle has much greater flexibility than the high ratio. This is because, with a suction nozzle, the air on the entering side remains at atmospheric pressure. Then, as the vacuum on the down-stream side increases, the air quantity that the nozzle will pass will increase only to the point of vacuum where the down-stream absolute pressure is about half the up-stream (atmospheric) pressure; 0.53 is the exact value. This would be reached with a vacuum of about 14.1 in. of mercury. For vacuums higher than this, the quantity of air passed by a given size high ratio nozzle would not increase. For each increase in air quantity, therefore, a new nozzle would be required. Furthermore, each high ratio nozzle would produce but one single vacuum for a given size pump.

Low Ratio Nozzle Test for Vacuum Pumps

It is obvious that if apparatus can be so arranged as to make it possible to use one nozzle for a wide range of vacuum points and a wide range of air quantities, a great convenience for test purposes will be secured. This can be done by using a low ratio nozzle as shown in Fig. 1, this being a test setup for a reciprocating pump. The various vacuum points desired are secured in a receiver *A* by adjusting globe valve *B,* which controls the amount of air drawn through receiver *C* and low ratio nozzle *D.* Several nozzle sizes may be used for the complete range of a test but a wide range of quantities may be handled for a correspondingly wide range of vacuum points in receiver *A,* with each size of low ratio nozzle. Furthermore, the quantity and, consequently, the vacuum in *A,* being controlled by adjustable valve *B,* instead of a fixed nozzle, it is possible to set the test points at any particular vacuum desired. It has been shown that if the vacuum is controlled by a high ratio nozzle, the vacuum attained is absolutely fixed by the nozzle size and not variable at will, except, of course, by a change of pump speed.

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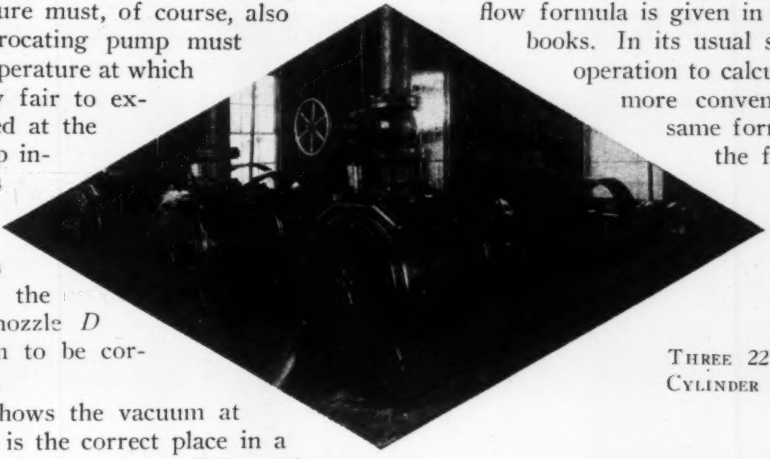
*Assistant Chief Engineer, Ingersoll-Rand Co., Easton, Pa., plant.

In measuring the air handled, this may be expressed in several ways: as cubic feet at atmospheric pressure, or as cubic feet at intake (vacuum) pressure or as pounds weight of air. In expressing the quantity as cubic feet, the temperature must, of course, also be stated. As a reciprocating pump must handle the air at the temperature at which it is received, it is only fair to express the volume handled at the temperature of the pump intake. In Fig. 1, this would be shown by thermometer *G*. Thermometer *H* is used only to calculate the flow volume through nozzle *D* and this volume is then to be corrected to temperature *G*.



Three 22 x 8 Belt Driven, Single Cylinder Vacuum Pumps in a Flotation Mill

Mercury column *E* shows the vacuum at the pump intake, which is the correct place in a test of the pump alone. Water column *F* is used to determine the air flow through nozzle *D*, a water column usually being sufficient for a low ratio nozzle.

Shape of the Nozzle

The next point to be discussed is the nozzle shape. Much research work has been done on this subject and it is quite well agreed that the nozzle with rounded entrance is the most accurate. It is practically impossible to produce an orifice in a thin plate with perfectly square entrance edges. In practice there is always some slight rounding of the edges, even if only to remove the burrs. The coefficient of flow is subject to rather wide variations when the edge is *almost* sharp. Thus, two nozzles supposed to have "perfectly sharp," or square edges are certain to be sufficiently different right at the edge to make a very appreciable difference in the flow quantity.

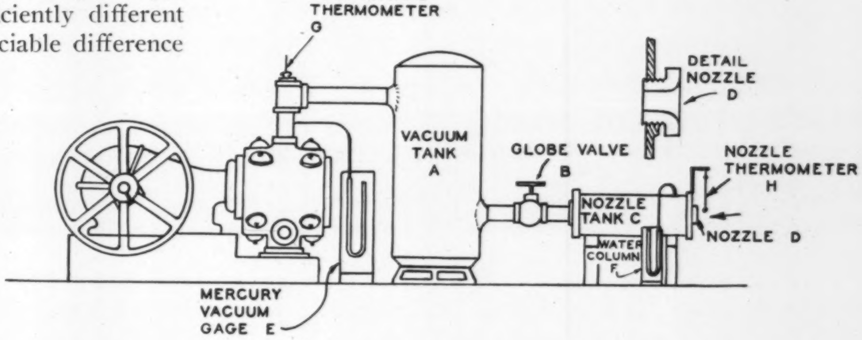


Fig. 1—Set Up for Nozzle Test of Dry Vacuum Pump

It has been found by research work that if a nozzle has well rounded approach surfaces, the coefficient of flow is remarkably constant over a wide range. As a result of this, this type of nozzle has become well recognized as an accurate instrument for the measurement of air. It is generally agreed that the radius of the entrance curvature shall be no less than 0.6 of the diameter of the nozzle throat. Also, to insure parallel flow through the nozzle, the length of the parallel section of the throat is made equal to the diameter of the throat. These are the minimum proportions, but the radius of the approach may be greater without appreciable change in the flow. A somewhat longer throat also probably will not affect the flow, but a length of one diameter is enough. It will be found, likewise, that an increase of approach radius produces a pretty large outside diameter for the nozzle when larger sizes are considered.

Figuring the Quantity of Air

To figure the quantity, the well known adiabatic flow formula is given in most engineering handbooks. In its usual shape, it is a staggering operation to calculate by it. A somewhat more convenient expression of the same formula as applied to *air* is the following:

$$W = \frac{2.04CAP}{\sqrt{T}} \sqrt{\left(\frac{1}{R}\right)^{1.425} - \left(\frac{1}{R}\right)^{1.712}}$$

In this formula:

W = weight of air discharged per second, pounds.

C = coefficient of flow. With a nozzle having well rounded approach surfaces and polished, it is from 0.97 to 0.99.

A = area of nozzle throat in square feet.

P = absolute pressure on up-stream side of nozzle, in lb. per sq. foot. For flow from atmosphere into a vacuum, P is absolute atmospheric pressure expressed as pounds per square foot.

T = absolute temperature in F on up-stream side of the nozzle (460 + thermometer reading).

R = ratio of absolute pressure on up-stream side of nozzle to absolute pressure on down-stream side of nozzle. This must be expressed to at least five significant figures.

After obtaining the weight of air flowing per second, it is usually necessary to express this in cubic feet at some stated pressure and temperature, usually the pressure and temperature of the vacuum pump inlet. This result is obtained by dividing the weight of flow per second, or per minute, by the specific weight of

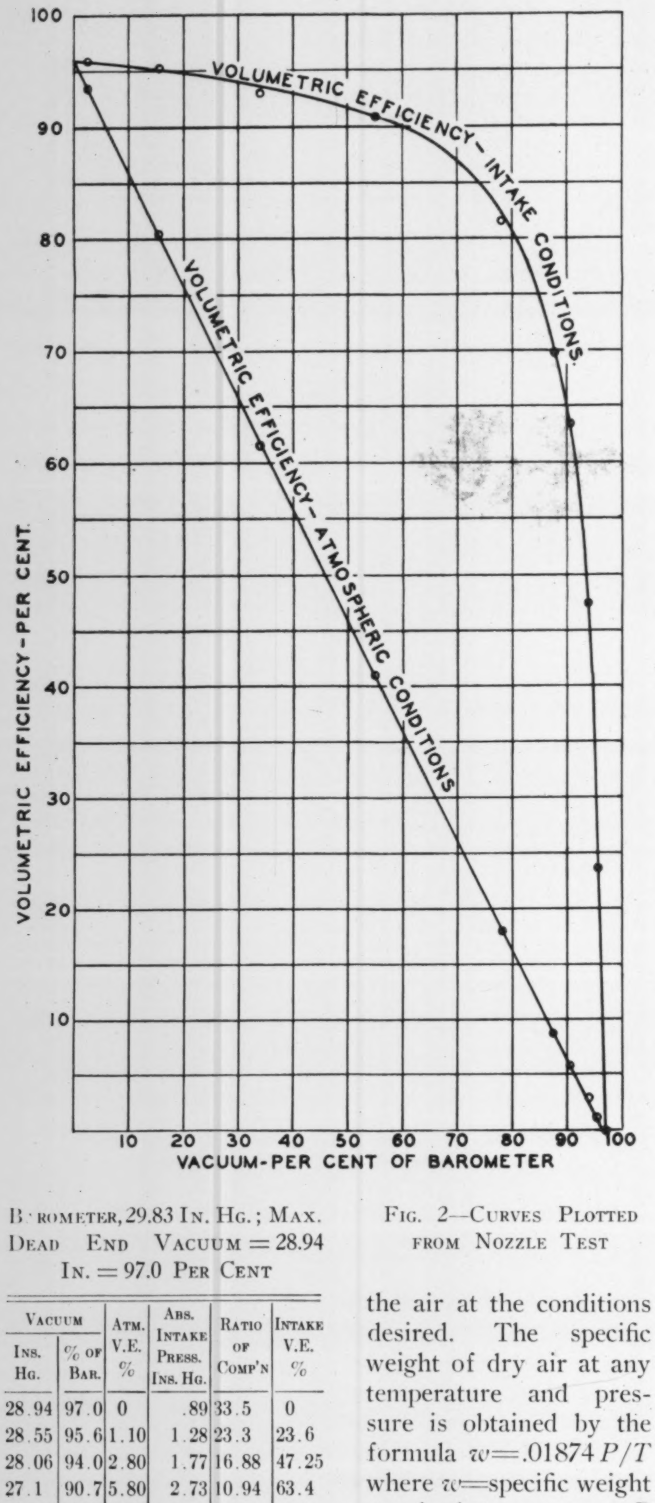


Fig. 2—Curves Plotted from Nozzle Test

Barometer, 29.83 In. Hg.; Max. Dead End Vacuum = 28.94 In. = 97.0 Per Cent

Vacuum		Atm. V.E. %	Abs. Intake Press. Ins. Hg.	Ratio of Comp'n	Intake V.E. %
Ins. Hg.	% of Bar.				
28.94	97.0	0	.89	33.5	0
28.55	95.6	1.10	1.28	23.3	23.6
28.06	94.0	2.80	1.77	16.88	47.25
27.1	90.7	5.80	2.73	10.94	63.4
26.1	87.5	8.70	3.73	8.01	69.7
23.3	78.0	17.8	6.53	4.58	81.5
16.43	55.1	40.8	13.4	2.23	90.9
10.14	34.0	61.4	19.69	1.516	93.0
4.65	15.6	80.5	25.18	1.186	95.4
.75	2.5	93.5	29.08	1.026	95.9

the air at the conditions desired. The specific weight of dry air at any temperature and pressure is obtained by the formula $w = .01874\,P/T$ where w=specific weight at absolute pressure P and absolute temperature T in pounds per cu. ft.

P = abs. pressure at which the specific weight is to be found, pounds per sq. ft.

T = abs. temp. in F at which the specific weight is to be found.

Formula for Slide Rule

The use of the flow formula given above is rather difficult. It is much easier, and with a nozzle flow pressure not exceeding 18 in. of water column almost as accurate, to use a formula which has been recognized for a number of years by numerous builders of air compressors and vacuum pumps. This formula, based upon simple hydraulic flow, is as follows and can easily be worked with a slide rule:

$$Q = 3.64\,C\,d^2 \sqrt{\frac{HT}{P_m}}$$

where

Q = cubic feet flowing per minute at absolute pressure of *down-stream* side of nozzle and flow temperature T.

C = coefficient of flow = 0.97 to 0.99, as before.

d = diameter of nozzle throat (smallest section) inches.

H = observed water column in inches (not over 18 inches).

T = absolute temperature in F, on up-stream side of nozzle.

P_m = absolute mean pressure; the average between pressures on up-stream and down-stream sides of nozzle, pounds per sq. inch.

As shown, the value of Q is expressed at temperature T and at pressure of the downstream side of the nozzle. In most cases, it is necessary to correct this value to some other condition of pressure and temperature so as to be able to compare quantities or to express the result as a *volumetric efficiency* as in the case of a reciprocating pump. In this latter case, Q should be recalculated to absolute pressure shown by the vacuum gage E and the machine intake thermometer G in Fig. 1. This air quantity divided by the piston displacement of the reciprocating pump gives the true volumetric efficiency of the pump under the observed conditions of the test.

Before going to the matter of rotary vacuum pumps, it would be well to work out an example with data from a reciprocating pump. Assume that in a test the valve B, Fig. 1, is adjusted to give a vacuum, E, of 26 in. Assume the barometer to be 29.18 in., and that a nozzle having a 3/8 in. throat diameter and well rounded and polished approach surfaces produces a water column, F, equal to 11.6 in. Let the room temperature, H, which is also the flow temperature in this case, be 64 F, and let the temperature of the pump inlet, G, be 58 F.

We have, assuming a flow coefficient of 0.98:

Up-stream pressure = atmosphere = $29.18 \times .491 = 14.33$ pounds per sq. inch absolute.

Flow pressure through nozzle = $11.6 \times .0361 = .419$ lb. per sq. in.

Down-stream pressure = $14.33 - .419 = 13.91$.

$$\text{Mean pressure} = \frac{14.33 + 13.91}{2} = 14.12 = P_m.$$

Also 64 F = $64 + 460 = 524$ deg. abs.

$$\text{Thus } Q = 3.64 \times .98 \times .375^2 \sqrt{\frac{11.6 \times 524}{14.12}}$$

$Q = 10.4$ cu. ft. per min.

This is expressed at down-stream pressure 13.91 lb.

per sq. inch, inside nozzle tank *C*, Fig. 1. It is also at flow temperature *H*, Fig. 1, which was given as 64 F. To get the true volumetric efficiency of the pump this quantity must be expressed at the conditions of the pump intake; that is, vacuum *E* and temperature *G*, Fig. 1. The vacuum was given as 26 in. of mercury and as the barometer was 29.18, the absolute intake pressure was 29.18 — 26.00 = 3.18 in. of mercury, or 3.18 × .491 = 1.561 lb. per sq. inch absolute. Also temperature of intake *G* was observed to be 58 F, or 58 + 460 = 518 deg. abs.

Then the flow volume corrected to conditions of pump intake in direct proportion to the absolute temperature and in indirect proportion to the absolute pressures, is

$$10.4 \times \frac{(58+460)}{(64+460)} \times \frac{13.91}{1.561} = \frac{10.4 \times 518}{524} \times \frac{13.91}{1.561} = 91.6 \text{ cu. ft.}$$

Let us further assume that the piston displacement of this pump at the observed speed is 139 cu. ft. per minute. Then the true *volumetric efficiency* referred to the pump intake is

$$\frac{91.6}{139} = 66 \text{ per cent.}$$

By varying the opening of globe valve *B*, Fig. 1, the vacuum at *E* may be varied and set at any number of values at which test figures are desired. The air quantities will be measured by nozzle *D* and numerous quantities may be measured by a given nozzle, as already explained, up to the limit set for the accuracy of the formula. Beyond this limit it will be necessary to change to another size nozzle.

Plotting Results

The intake volumetric efficiency points thus obtained may be plotted against per cent vacuum as shown by the upper, or curved line in Fig. 2, marked "Volumetric Efficiency at Intake Conditions." It will be observed that as the vacuum increases, the air quantity and so the volumetric efficiency decreases when we finally reach the condition that globe valve *B* in Fig. 1 is completely closed, no air at all is taken in by the pump. This is the maximum or, "dead end vacuum" that the pump will pull and the volumetric efficiency is zero.

Atmospheric Volumetric Efficiency

In Fig. 2, a diagonal line marked "Volumetric Efficiency Atmospheric Conditions" will be observed. This is an exceedingly useful curve for estimation purposes; by its use the approximate performance of any reciprocating pump may be estimated very easily and the curve itself may be approximated merely by observing the dead end vacuum that any given pump will pull. Thus, for very many rough checks, only the dead end vacuum need be determined.

The atmospheric volumetric efficiency is determined by expressing the air handled by the pump at atmospheric pressure, instead of at intake (vacuum) pressure, and then dividing this by the piston displacement. For instance, taking the problem already figured, instead of correcting the air flow quantity to the conditions of pump intake to get the true volumetric efficiency, we can correct the flow volume to the pressure of the atmosphere. We must still refer it to the temperature of the pump intake (thermometer *G*) because the air enters the pump at that temperature. The nozzle gave us 10.4 cu. ft. per minute at the pressure of the downstream side of nozzle, 13.91 lb. per sq. inch, and flow temperature 64 F. Correcting this to atmospheric pressure, 14.33 lb. per sq. inch and intake temperature *G*, which is 58 F, in inverse proportion to absolute pressures and direct proportion to absolute temperatures, we have

$$10.4 \times \frac{13.91}{14.33} \times \frac{(58+460)}{(64+460)} = 10 \text{ cu. ft. per minute.}$$

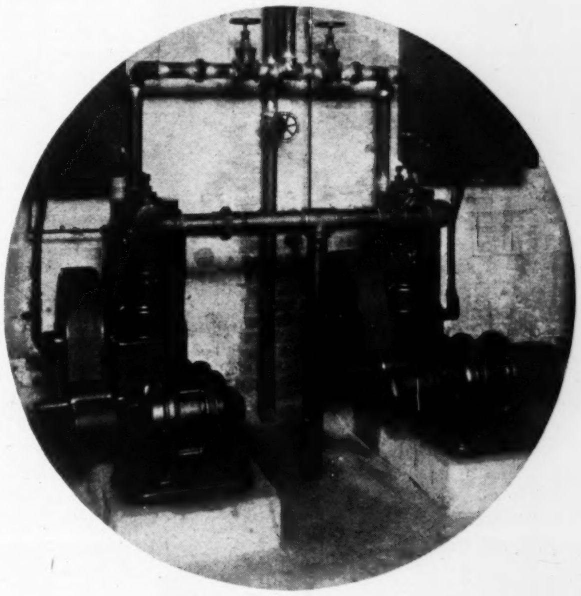
Dividing this quantity by the piston displacement per minute gives

$$\frac{10}{139} = 7.2 \text{ per cent.}$$

This is the volumetric efficiency with air volume expressed at atmospheric pressure.

To make the comparison clearer, we repeat the definitions of the two kinds of "volumetric efficiencies:"

True, or Intake Volumetric Efficiency. This is the ratio of the actual volume of air handled in a given time to the piston displacement developed by the pump during this time. The air volume is expressed at the conditions of temperature and pressure (vacuum) existing at the intake connection of the pump cylinder.



Two Vertical, Belt Driven Vacuum Pumps in a Disposal Plant

Atmospheric Volumetric Efficiency. This also is the ratio of actual volume of air handled to piston displacement and the air volume is expressed at the temperature of the intake connection of the pump cylinder; but at *atmospheric pressure.*

It is obvious that if the volumetric efficiencies found from a series of tests are plotted against vacuum, we should find the points to form smooth curves. The usual procedure is to let horizontal distances represent vacuum and the ordinates represent volumetric efficiencies. Because the barometer varies, it is customary to express the vacuum as percentage of the barometer reading. For instance, if the test vacuum point was at 26.6 and the barometer is 29.83 the vacuum would be expressed as

$$\frac{26.6}{29.83} \times 100 = 89.2 \text{ per cent}$$

Fig. 2 shows how a properly run test will look when the results are plotted. The upper curved line is the true or *intake volumetric efficiency* and the diagonal, almost straight, line shows the *atmospheric volumetric efficiency*. On each curve, at zero vacuum (pumping from atmosphere to atmosphere) the volumetric efficiency is the same and should be well above 90 per cent. A fairly good pump with automatic valves should show around 96 per cent or better, because, except for slight pressure losses through the machine, the pump cylinder is acting practically as a meter. Incidentally this is rather convincing evidence of the accuracy of the nozzle method of air measurement.

It will further be observed that both volumetric efficiency curves also come together at the point of maximum vacuum. This point is reached when the intake is completely shut off (dead end) and no air at all is passing through the pump. As has been mentioned before, at this point, the volumetric efficiency, no matter how the air volume is expressed, is zero.

Approximate Performance Determinations—Practical Uses

Now we come to a very convenient use which may be made of the *atmospheric volumetric efficiency* line and a method of estimating the approximate performance of a pump with but one test point. It will be observed that the *atmospheric volumetric efficiency* line is very nearly a straight line between zero efficiency at maximum, dead end vacuum and about 96 per cent volumetric efficiency at zero vacuum. There appears always to be a slight sag in this line toward the higher vacuum end. This sag seems to be caused by machine valve leakage.

If, therefore, it is desired to find out about what air volume a given pump will handle at a given vacuum, it is necessary only to close the intake and measure the maximum, dead end vacuum compared to the barometer. In this test, of course, every leak must be closed because even a very small leak will reduce the maximum vacuum attained. If there is any appreciable length of intake pipe back to a shut-off valve, the pipe joints should be painted with some kind of thick dope, as is usual with such piping. Also something will doubtless have to be done to the stuffing box of the shut-off valve, as it will usually show a leak on such a test.

Having obtained the maximum vacuum in per cent of the barometer, a diagonal line may be drawn between zero volumetric efficiency at this maximum vacuum and a point showing 96 per cent volumetric efficiency at zero vacuum. Put a little "sag" into this diagonal line, toward the high vacuum end so as to resemble Fig. 2, and the result is a very fair approximation of the air handling performance of the pump. Of course, it is not a true measure but the values so obtained may be used for various purposes in a plant. (The amount of sag put into the line must be a matter of judgment as to whether the pump valves are in good or bad condition.)

One problem that may be solved by the use of this curve is to get an approximation of the air leakage into a given vacuum exhaust system. With the observed vacuum, the pump speed, the approximate volumetric efficiency of the pump obtained as just described and the temperature of the mixture in the system, the principle of partial pressures may be used to determine how much steam and how much air is in the mixture. You will remember that with such a mixture, the air and the steam, each, occupy the entire volume handled and each is at its own partial pressure. The pressure of the steam corresponds to its temperature (from steam tables) and the partial pressure of the air is the difference between the total observed pressure (vacuum) and the pressure of the steam. The air quantity is, therefore, the entire volume handled at observed temperature and at its own partial pressure thus obtained.

Similarly, a test may be made on a true dry vacuum system handling only air. Having determined the dead end vacuum that the pump alone will pull, say with a blank flange inserted at the pump intake, and then observing the vacuum that can be obtained with pump connected to the vacuum system, the air quantity may be obtained. If in this test, all air supply is cut off, the measurement will show the leakage of the system.

These are but two possible and useful problems that may be solved approximately by the method described.

Thus far we have considered constructing only the approximate atmospheric volumetric efficiency line. The approximate *intake volumetric efficiency* curve may be obtained from this merely by multiplying the values on the atmospheric volumetric efficiency line by the ratio of compression. It has been explained that the intake volumetric line expresses the air volume at intake pressure, which is the vacuum, and the atmospheric volumetric line expresses the air volume at atmospheric pressure. Both lines express the volume at the same temperature; that is, the temperature of the intake connection of the pump. It, therefore, follows that the volumes expressed by the two lines are in the inverse ratio of the pressures. Thus, the intake volume may be obtained by multiplying the atmospheric volume by the ratio of the absolute atmospheric pressure to the absolute intake pressure. The absolute intake pressure is the difference between the barometric pressure and the observed vacuum, expressed either as pounds per square inch or inches of mercury. Thus with barometer at 29.86 in. and the vacuum at 26.43 in., we may figure the atmospheric pressure and also the vacuum in pounds per square inch, or subtract the vacuum reading from the barometer, giving 3.43 in. mercury absolute intake pressure. As the atmosphere is at 29.86 in., the ratio of pressures and ratio of volumes is

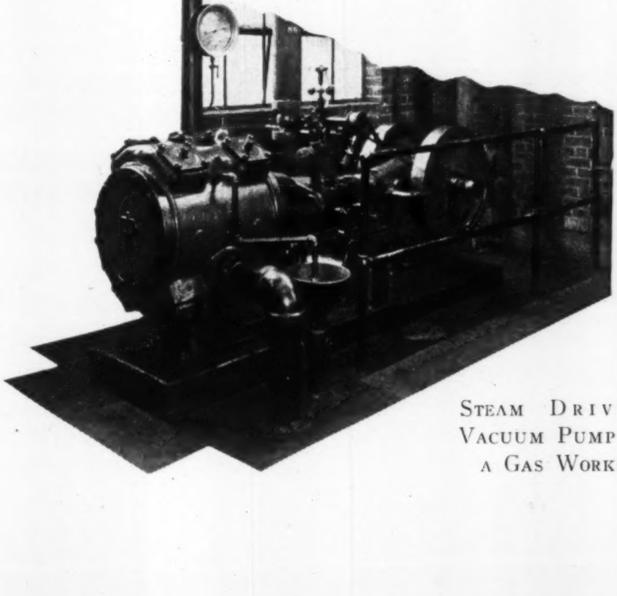
$$\frac{29.86}{3.43} \text{ or } 8.71$$

Thus, the intake volume is 8.71 times the same quantity of air expressed at atmospheric pressure and, therefore, the intake volumetric efficiency at 26.43 in. observed vacuum (with 29.86 in. barometer) is 8.71 times the *atmospheric volumetric efficiency.* Also it will be observed that if the pump is discharging into the atmosphere, 8.71 is the *ratio of compression* of the pump; that is, the ratio of the discharge pressure to the intake pressure, both absolute. Therefore, the approximate *intake volumetric efficiency* curve may be constructed from the approximate *atmospheric volumetric efficiency* line by multiplying the values on the latter line by the absolute ratio of atmospheric pressure to the aboslute pressure of the observed vacuum, which ratio, when the pump is discharging at atmoshpheric pressure, is also the ratio of compression for each point. Of course if either curve is obtained from actual test, the other curve may be calculated from the first one by the same method and both will be the true test values, subject only to the errors of test.

Referring to Fig. 2, the table shows how the true, or intake volumetric, efficiency curve may be calculated from the points on the atmospheric volumetric efficiency line. This curve also shows that because of the steepness of the intake volumetric line, the values, say from 90 per cent vacuum up, are very indeterminate. The slightest shift to right or left makes an enormous difference in the values of intake volumetric efficiency.

This is not the case with the diagonal atmospheric volumetric efficiency line; the values on this line are easily read throughout its length. Therefore, in calculations based upon tests so plotted, it is better to refer to the atmospheric volumetric line. These values for calculation purposes may be changed to intake volumetric values by the method shown in this text and also in the tabulation in Fig. 2. It must be understood, however, that the ratios of compression in the table, Fig. 2, apply only to the barometer observed in that particular test as shown. For any other test, the ratios must be figured from the actually observed barometer reading.

We have said that the ratios of the values on the two volumetric lines correspond to the ratio of compression for each test point when the pump discharges at atmosphere. In case the pump discharges at some other pressure, either below or above atmosphere, the calculation is the same; that is, the ratio for each point is the ratio of the observed *atmospheric* pressure to the absolute pressure of the vacuum. The actual pump discharge pressure will not affect this part of the calculation; but in this case the ratio will not be the ratio of compression. The ratios between the points on the two curves are always merely the *ratios of the two pressures at which the values are to be expressed* and plotted for each point.



Steam Driven Vacuum Pump in a Gas Works

Another matter to be observed in plotting the intake volumetric line from the atmoshperic line is that any experimental errors on the atmospheric line are exaggerated when re-plotted on the intake line. In fact such errors are in each case multiplied by the ratio of pressures for each particular point. It is better, therefore, to plot the intake line through points calculated from "smooth curve" values taken from the plotted atmospheric line itself, instead of from the actual test points. This will give much smoother points from which to plot the intake volumetric curve. It will be a great help to plot in a few extra points where the intake curve makes the rather sudden downward turn, as between 60 and 85 per cent vacuum, in Fig. 2. Of course it is well to plot the actual test points along the intake line afterwards and it should show that the curve as plotted through the smooth curve points also consistently agrees with the re-plotted actual test points, though these latter points will be distributed on each side of the smooth curve.

Rotary Pumps-Hurling Water Type

The question has been asked as to how to test a rotary vacuum pump of the "hurling water" type. With such a machine the test procedure would be the same as for the reciprocating pump except the matter of the temperature at which the volume of air handled would be expressed. With this kind of a pump, the test result is the amount of air actually handled in a given time and not a volumetric efficiency; but in order to compare the performances of two pumps, the air volumes in the two cases must be expressed at the same temperature. Of course, also, the volumes must be expressed at the same pressure in each case. The pressure may be that of the pump intake (vacuum) or, if one pump pulls a higher vacuum than the other, it would be logical to express the air volume at atmospheric pressure in both cases.

In a hurling water type of pump, the water volume is so large compared to the air volume that we can assume, for all practical purposes, that the air will at once be brought to the same temperature as the incoming water. As a matter of fact the temperature of the mixture will depend upon the relative amounts (weights) of air and water handled in a given time, but as said, for all practical purposes, the air temperature will not appreciably affect the water temperature.

One might suggest expressing the air volume at the observed temperature of the mixture discharged, but this will be influenced by the heat of compression of the air and also the losses in the pump. Altogether it would seem best to express the air volume at the temperature of the intake water. In comparing two pumps it is imperative that the hurling water to each pump must be at the same temperature. If this is not the case the results are not directly comparable; first because the air volumes will be expressed at different temperatures and second, because the colder the water, the more air (weight) will be handled and so the comparison would be unfair. It is, therefore, not enough in such a case merely to *calculate* the air volume back to the same temperature condition; both pumps on test must be supplied with water at the same temperature, because this water temperature will in itself influence the performance.

Predicting the Flow of Liquids in Pipes

By R. E. Gould

THROUGH the use of Rayleigh's[1] principle of dynamical similarity, various experimenters have found it possible to correlate data on the flow of liquids in pipes in such a way that the performance of any liquid flowing in a pipe under any conditions may be readily and accurately predicted.

A method of predicting the friction pressure loss which is coincident with the flow of any fluid through a pipe line is herein discussed, and the solution of this problem for several liquids, pipe sizes, etc., is presented in graphical form. In addition, the effect of some of the variables such as temperature and specific gravity on the pressure loss are discussed briefly.

Of the two main types of flow existing, viscous and turbulent, that encountered in most engineering practice is of the eddying, unordered class, and accordingly this treatment is confined to liquids in turbulent motion.

The pressure loss in a pipe line flowing full of liquid under turbulent conditions is expressed by the Darcy formula

$$p_1 - p_2 = \mathrm{f}\,\frac{1.01lM^2}{10^7\, d^5\, gs} \qquad (1)$$

in which

$p_1 - p_2$ = pressure loss due to friction, lb. per sq. in.
f = friction factor
l = effective length of pipe, ft.
M = rate of circulation of liquid, lb. per min.
d = inside diameter of pipe, ft.
s = specific gravity of liquid
g = acceleration of gravity, ft. per sec. per sec.

In order to evaluate the pressure loss, $p_1 - p_2$, by using Equation (1), a knowledge of the magnitude of the friction factor, f, is essential. It has been shown[2] that for pipes having the same relative degree of roughness on their inner surfaces there exists for all liquids flowing in turbulent motion a single mathematical relation between the factor f and a dimensionless ratio known as Reynolds' number. This ratio may be defined by the following equation

$$R = d\,v\,k \qquad (2)$$

in which

d = inside diameter of pipe, ft.
v = average liquid velocity, ft. per sec.
k = kinematic fluidity, sec. per sq. ft.
(This variable k is sometimes stated as the ratio between the density and absolute viscosity)

A graphical representation of the relation between the friction factor, f, and Reynolds' number, R, for wrought iron pipe is shown in Fig. 1[3].

The problem of determining the pressure loss for a given set of conditions resolves itself into the following steps:

1. Determination of the Reynolds' number by substituting proper values in Equation (2).
2. Obtaining the friction factor, f, corresponding to this Reynolds' number, from Fig. 1.
3. Calculating the pressure loss by substitution in Equation (1).

As an example let it be desired to find the pressure loss occurring in a 1¼-inch pipe 250 feet long, delivering 300 lb. of 40 F water per minute.

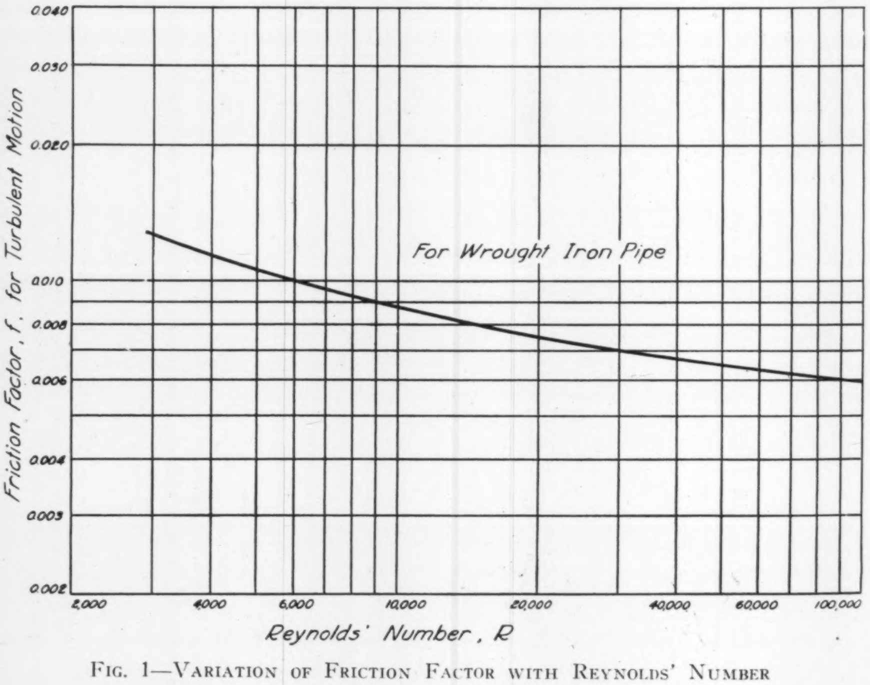


FIG. 1—VARIATION OF FRICTION FACTOR WITH REYNOLDS' NUMBER

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[1] Lord Rayleigh, Phil. Mag., Vol. 34, 1892.
Lord Rayleigh, Phil. Mag., Vol. 8, 1904.

[2] Flow of Brine in Pipes, by R. E. Gould and M. L. Levy, Univ. of Ill. Eng. Exp. Sta. Bulletin No. 182.

[3] The data for Figs. 1 and 4 were obtained from an investigation conducted in the Mechanical Engineering Dept. of the University of Illinois by the Engineering Experiment Station.

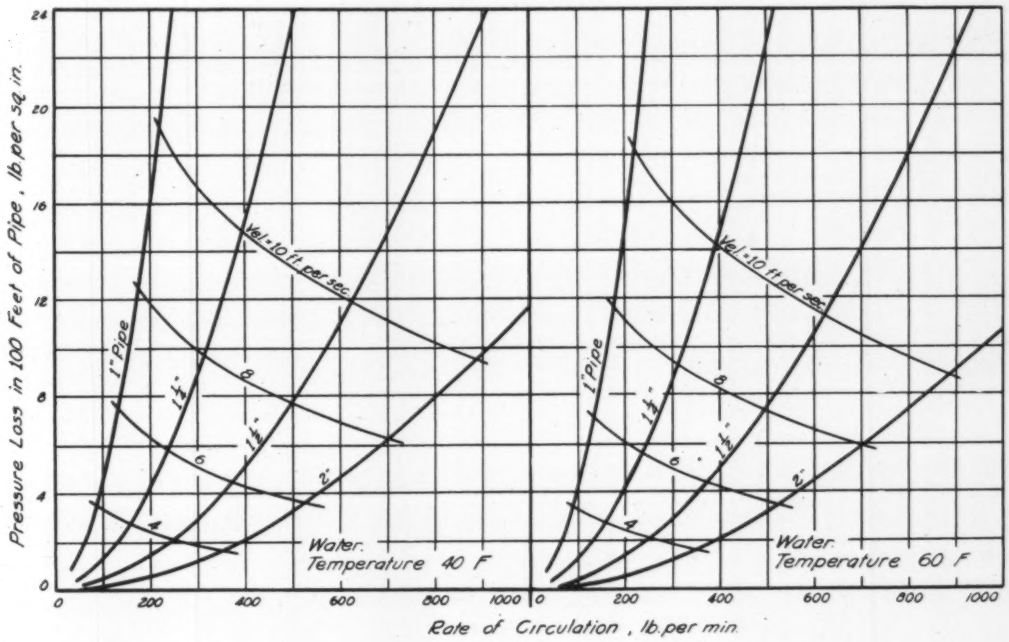

FIG. 2—CIRCULATION OF WATER IN WROUGHT IRON PIPE

Determination of Reynolds' number.

$$R = d\,v\,k$$

in which

$$d = \frac{1.38}{12} = 0.115 \text{ feet}$$

$$v = \frac{300}{60 \times 62.4 \times \frac{\pi}{4}(0.115)^2} = 7.7 \text{ ft. per sec.}$$

For 40 F water ($s = 1$) from Fig. 4

$k = 60{,}000$ sec. per sq. ft.

$\therefore R = 0.115 \times 7.7 \times 60{,}000 = 53{,}100$

Obtaining the friction factor:

From Fig. 1, the value of $f = 0.0063$ corresponds to $R = 53{,}100$

Calculation of pressure loss:

Substitution of the predetermined values in Equation (1) gives

$$p_1 - p_2 = 0.0063 \frac{1.01 \times 250 \times 300^2}{10^7 \times (0.115)^5 \times 32.2 \times 1} = 22 \text{ lb.}$$

per sq. in.

A number of cases involving the circulation of both water and commercial calcium chloride brine in wrought iron pipes, have been calculated by the above method and are shown in graphical form in Figs. 2 and 3. Similar curves for any liquid may be constructed provided the

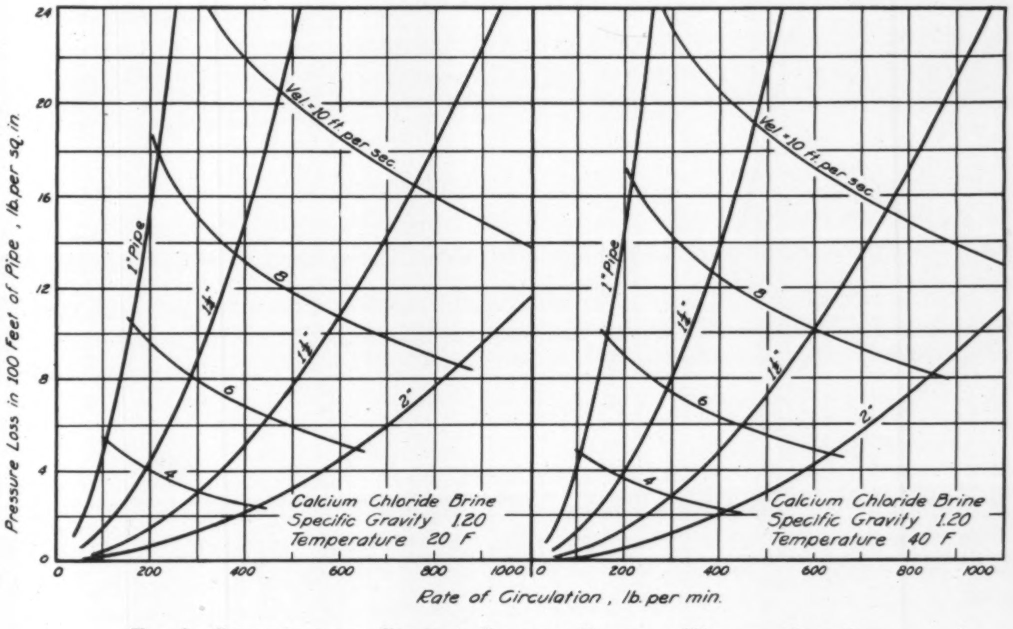

FIG. 3—CIRCULATION OF CALCIUM CHLORIDE BRINE IN WROUGHT IRON PIPE

data on the fluidity of the particular liquid are available. In Fig. 4 the relation of the kinematic fluidity of water and calcium chloride brine with temperature for various specific gravities is given. The values of k used in Equation (2) for computing Figs. 2 and 3 were obtained from Fig. 4.

It may be noted from Figs. 2 and 3 that in a given pipe the pressure loss increases very rapidly with increasing rates of circulation. Also with a constant rate of circulation the pressure loss increases rapidly as the size of pipe is decreased.

The effect of changes in temperature of the liquid on the pressure loss is shown in the left half of Fig. 5. A slight decrease in pressure loss is produced by an increase in temperature. This is, of course, readily attributed to the fact that the liquid is less viscous at higher temperatures. In the right half of Fig. 5 it may be observed that a small decrease in pressure loss accompanies an increase in specific gravity while maintaining a constant rate of circulation, i.e., a given number of lb. per min. While such an effect is rather difficult to visualize, it should be noted that an increase in specific gravity under such conditions means a decrease in the actual velocity and hence a decrease in pressure loss is realized.

Conclusions

From the foregoing it is evident that in order to reduce the friction pressure loss in any contemplated installation involving the circulation of a definite number of pounds of liquid per unit of time it is essential to circulate the liquid at the highest permissible temperature with the lowest possible specific gravity and at a low velocity. It is believed that the method of calculating the friction pressure loss, and the scheme of presentation of the losses in various sizes of pipe illustrated in Figs. 2 and 3 tends to place the selection of the most economical size of pipe for a particular installation on an accurate and rational basis.

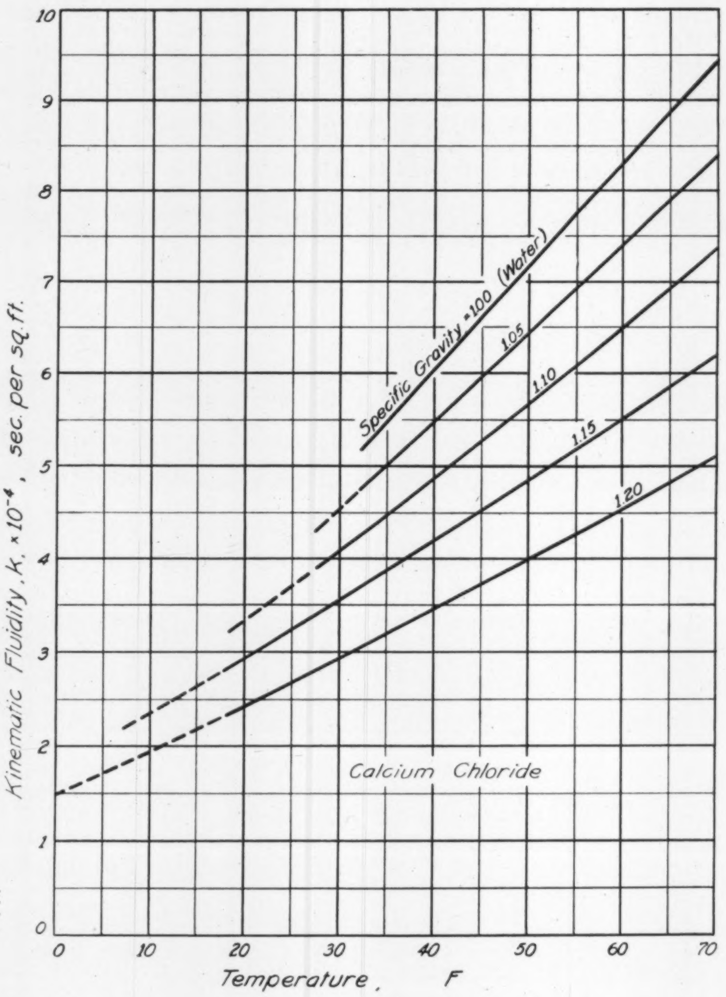

Fig. 4—Kinematic Fluidity and Temperature

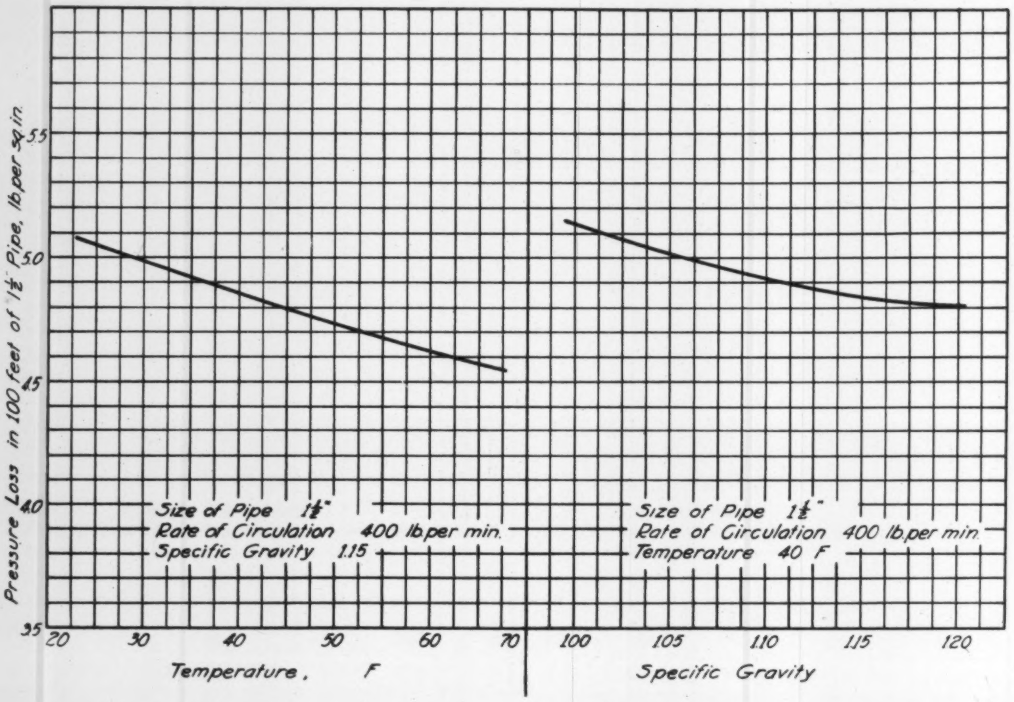

Fig. 5—Effect of Changes in Temperature and Specific Gravity on Pressure Loss

Heating·Piping
and Air Conditioning

1895

American Society *of* Heating *and* Ventilating Engineers
JOURNAL SECTION

OCTOBER *1930*

Material in this Section prepared by Publication Committee: G. L. Larson, *Chairman;*
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Welded Piping for Building Heating Systems

By F. G. Outcalt[1], Chicago, Ill.
NON-MEMBER

NEW developments in the heating industry have led to increased efficiencies and economies in operation. In the modern systems of steam and hot water heating practically every item involved has been changed in some manner, with the notable exception of the piping. Until recently, lengths of pipe have been joined by cutting to length and threading. Now, welding is being used to a large extent for the fabrication of piping systems for heating, though pipe welding is by no means new, as it has been used in various industries for many years, particularly in the petroleum and chemical industries.

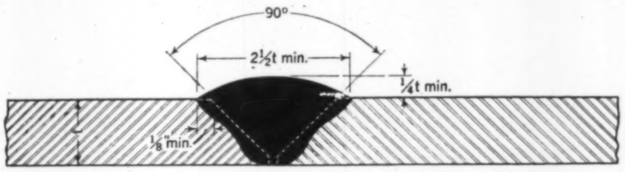

Fig. 1—Open Single Vee Butt Weld

Efficiency of Welded Construction

The term *efficiency* as used in this paper is intended to signify the degree of freedom from objectionable interferences with or the effects upon the purpose for which the piping system is installed, and involves the fittings, valves, or other appurtenances which might have an effect upon the flow of gas or fluid through the system.

Restrictions against the freedom of the flow of fluid or gas are eliminated where welded joints are used, and such welded joints are permanently tight, so that leakage losses are avoided. The danger of corrosion and leakage is greatly reduced, because corrosion does not take place at the welded joint. Furthermore, the welded joint is as strong as the pipe itself, and there is no necessity for increasing the pipe wall thickness to provide for the necessary strength at the roots of threading.

Economy of Welded Construction

The economy of the welded piping system begins in the drafting room, due to the simplicity of welded construction. Standard pipe for the entire system may be purchased in single or double random lengths, depending upon the nature of the job. Material costs are lowered through the elimination of fittings since, with the exception of valves and their connecting flanges, these are made up on the job from standard pipe. This is particularly important in the case of headers, elbows, bends, anchorages and supports, where all fittings may be eliminated except at the valves. In some recent installations, valves have been furnished with short nipples welded to the valve body, so that no flanges were required, the valves being welded directly into the line. A saving in erection is next effected through the reduction of total weight of pipe and supports and the consequent cost of handling. Much scaffolding is eliminated since sections of the pipe may be fabricated on the floor and tied into the system at convenient points. The cost of insulating the pipe is lowered as there are no flanges or flanged fittings to be covered with molded insulation. Waste material is almost entirely eliminated because short lengths of pipe can be used and no extra cast fittings are required. No delays are experienced because of alterations or additions which are not in the original plans, or in waiting for special fittings to arrive from the factory.

Joint Designs

The open single *vee butt* weld, shown in Fig. 1, is the standard welded line joint. It is easily and economically made and according to tests of many types of joints, it appears to be the strongest. Pipe to be joined by this weld is supplied by the manufacturer with ends machine-beveled, as shown in Fig. 2. The bevel does not extend to the inner wall of the pipe; a flat portion about 1/16 in. wide, for all sizes of pipe, is left to facilitate weld depth penetration and lining up. It also serves to strengthen the edge during transit and handling. When pipe is lined up for welding, a space *S* should be left between pipe ends to allow them to draw together as the cooling weld metal contracts, thus eliminating the possibility of overlapping edges or undue residual stresses. No allowance need be made for this spacing in design layout of a piping system as the overall lengths will be true within construction tolerances after welding. To add an extra factor of safety, the weld is usually built up, or reinforced, about one-fourth of the wall thickness, as shown in Fig. 1. This reinforcement should slope gradually from the center down to the surface of the pipe alongside the weld. The width of the weld should ordinarily be about two and one-half times the thickness of the pipe wall. This type of weld is recommended for

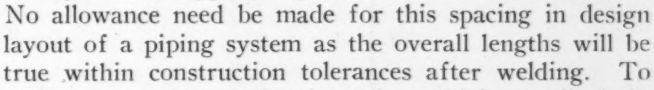

Fig. 2—Pipe Beveled for Welding

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[1] Resident Development Engineer, The Linde Air Products Co.

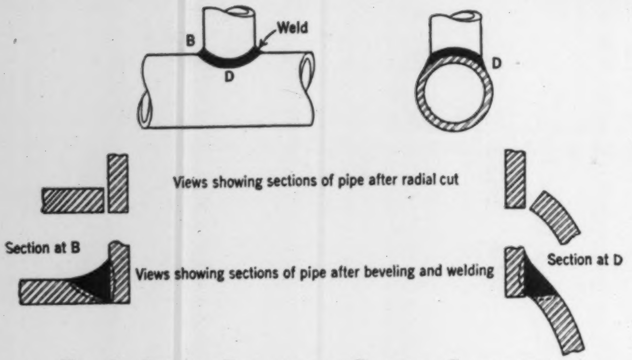

Weld
B
D
D
Views showing sections of pipe after radial cut
Section at B
Views showing sections of pipe after beveling and welding
Section at D

Fig. 3—Design Details for Branch Connection

standard, extra heavy and double extra heavy piping, for all services, carrying pressures to which steel or wrought-iron pipe is subjected.

Tee fittings, crosses, and branch connections, either of equal or unequal diameter, can be fabricated from standard pipe by cutting with the blowpipe and welding to form the finished fitting. Templets are used for making the cuts accurately. Such fittings should be carefully prepared to permit proper fusion to the bottom of the vee and to present a smooth flow condition. One of the most desirable designs is obtained by inserting the branch into the hole down to the inner surface of the header wall. When the cut in the header has been properly beveled, this type of intersection presents a very good vee for welding. Fig. 3 illustrates this type of assembly. These two types of joint design are fundamental and are most frequently used in welded piping construction, although other designs are used occasionally for various fittings.

Fittings for Welded Piping

With recent rapid developments in the welding of pressure piping, there have been introduced a number of novel features which not only make for increased speed of erection, but also improve the strength, continuity and efficiency of the finished piping system. These include the forged high-hub welding flanges and forged reducing fittings. Forged high-hub welding flanges are simply forged steel flanges having a beveled hub of sufficient length so that they can be welded to the end of a pipe without any possibility of distorting the face of the flange. Forged reducing fittings are short lengths of pipe swaged down at one end, which can be welded into the line at any point where a change in diameter is required.

Another class of fittings designed especially for welded construction is the short-radius quarter and 180-deg. elbow. These short radius elbows are forged from seamless steel pipe or tubing by special processes, and are of great service to welding erectors on account of the ease with which a section can be cut off and welded in to form an elbow of any desired angle. They have ample strength, are sightly in exposed piping, and greatly facilitate the application of insulating covering.

Fig. 4 shows the use of 12-in. extra heavy short radius bends in an installation designed for a working pressure of 200 lb per square inch.

Welded Headers

Fig. 5 shows a typical welded pipe header fabricated by means of the types of joints previously shown. Its counterpart constructed with standard cast iron flanged tees is also shown. The welding process is particularly adaptable to this type of piping construction. Headers are entirely fabricated from standard pipe, without the use of cast or forged fittings, unless flanges are used for connecting to branch piping or valves. Thus a compact design is obtained with great economy of weight and expense. Trouble from leaky gaskets is permanently eliminated. Delay in waiting for delivery of fittings, which

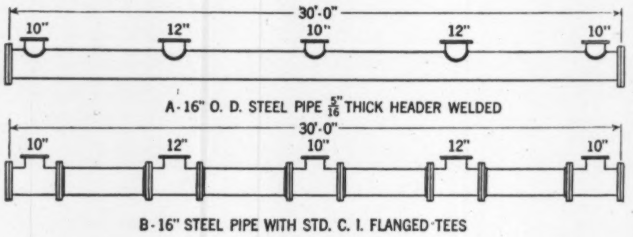

30'-0"
10"
12"
10"
12"
10"
A-16" O. D. STEEL PIPE 5/16" THICK HEADER WELDED
30'-0"
10"
12"
10"
12"
10"
B-16" STEEL PIPE WITH STD. C. I. FLANGED TEES

Fig. 5—Header Designs

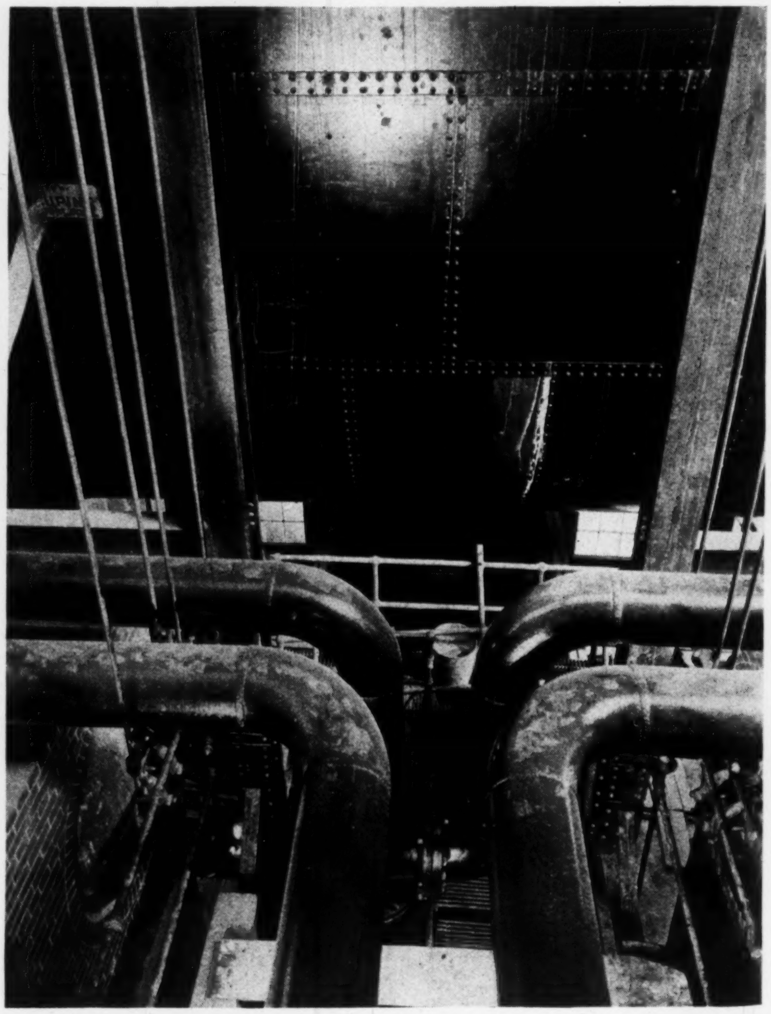


Fig. 4—Short Radius Elbows in 12 In. Extra Heavy Line

sometimes cannot be ordered until boilers and other equipment are in place and lined up, is avoided. Due to the flexibility of welding processes, minor changes in design not included in the original plans can easily be made. Insulation can be applied neatly and at low cost to a welded header, due to the elimination of flanges. A welded header weighs much less than a similar one made up of cast fittings.

Another type of header, of more recent design, which has some very interesting features, is shown in Figs. 6 and 7. This is the main header for four 250-hp boilers for heating a large building containing approximately 50,000 sq ft of direct radiation, augmented by several unit heaters and a central blast system. This header consists of two 16-ft lengths of 16-in. pipe welded together and blanked off at each end by pieces of ¾-in. flat plate welded in. The four 10-in. boiler leads are welded throughout from the long-hub, forged-steel welding flange and short radius bend at the boiler connection to the short radius bends welded to the header at the other end. All of the mains are taken off by means of short radius bends. These are welded directly into the header and perform double duty in providing outlets for the header and changes in direction through the use of a single fitting. The entire header, including all connections, was fabricated on the floor and hoisted into position in one piece. The space saving is notable, the overall vertical dimensions being about 34 in., which compare with a minimum of 53½ in. required for flanged construction. The weight of the welded header is approximately 2,650 lb while the weight of a similar header made up of flanged cast-iron fittings would be about 10,400 lb or nearly four times as great. Another 6-in. outlet was later welded into this header.

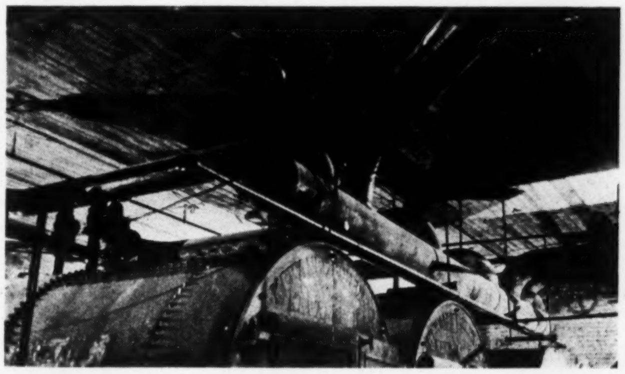


Fig. 6—Novel Header Design

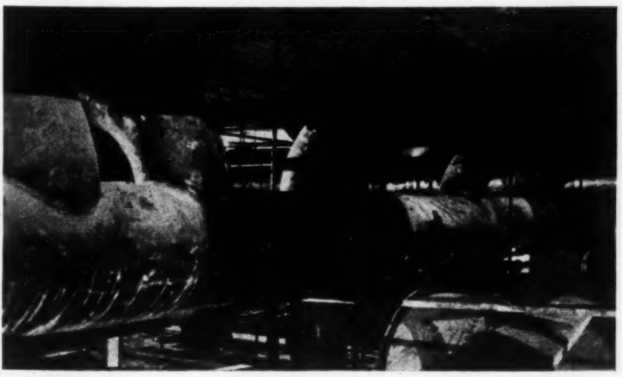


Fig. 7—Short Radius Bends Welded Direct to Header

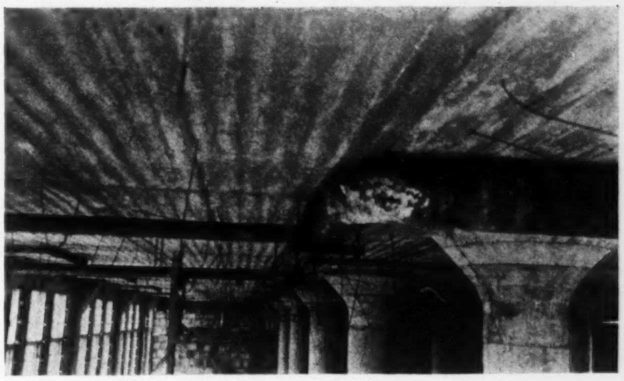


Fig. 8—Welded Piping Installed in Minimum Space

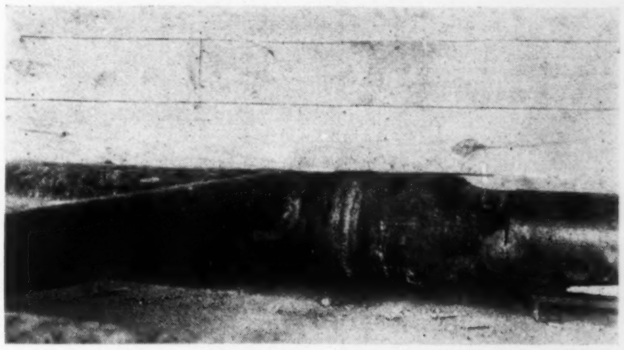


Fig. 9—Welded Reducer and Branch Connection

Adaptability of the Oxy-Acetylene Process

The adaptability of welded construction to structural details of the building in which it is installed is illustrated by the following change of plans in the distribution floor that occurred during the construction of a large building, a view of which is shown in Fig. 8. As originally planned, the mains were to be exposed, and it is evident from the 16½ in. space above the tops of the windows that there is no room for an attic of the usual proportions for concealing the piping. Since this floor was to be used for offices, it was later decided that exposed piping was not desirable and that the pipe should be concealed if possible. The main and branches were successfully placed in a space of 14¾ in., after allowing for steel and lath and plaster for the hanging ceiling below the pipe. Since the largest main was 12 in. in diameter, leaving only about 1-in. clearance above and below the pipe, it was necessary that runouts to the branch risers be welded into the main eccentrically to provide for drainage of condensate. To insure accuracy, templets were used in making the cuts for these connections. This main was fabricated on the floor in lengths of 40 to 80 ft with short lengths of branches welded in.

The sections were raised and hung temporarily a few inches below their final positions by the use of long hanger rods. Space was thus provided for making the position welds and the main was later raised to its permanent location by screwing up the hanger nuts, after which the surplus rod was cut off. Fig. 8 shows the proximity of the piping to the ceiling. Fig. 9 shows the details of a reducer and branch connection in this main on the floor before it was hoisted into position. The

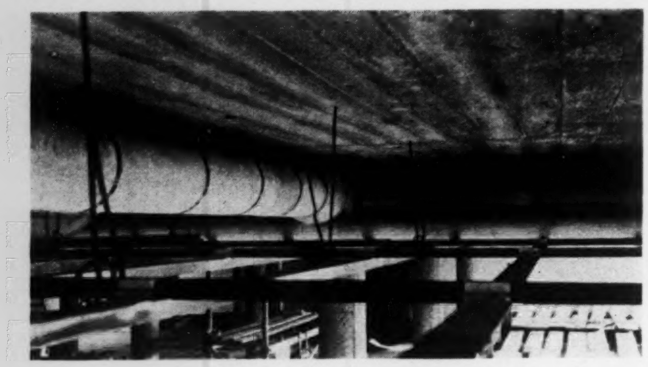


FIG. 10—WELDED PIPING COVERED WITH INSULATION AND READY FOR THE HANGING CEILING

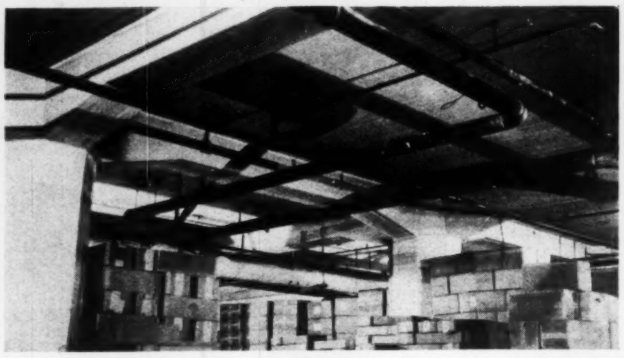


FIG. 11—WELDED EXPANSION LOOP

standard 45-deg bevel and the spacing of the pipe ends to permit thorough fusion and to provide for contraction of the cooling weld metal, are plainly shown. The appearance of this main after insulation was applied and ceiling steel was in place before lathing is shown in Fig. 10. A 6-in. main is shown in the foreground and the reduction from the 8-in. main, after a branch has been taken off, is shown in about the center of the illustration. The latter main was installed level on account of the small amount of space available, and all condensate was carried off by the eccentric branches.

Expansion and Contraction

Provision for expansion and contraction of welded piping is made in practically the same way as for other types of piping construction. Standard expansion loops are ordinarily used. Good practice in installing expansion loops is to spread them a sufficient amount by means of a jack, prior to connecting, so that they will be in a neutral position and not subject to any bending stress when the line is at operating temperature, if the line is to be hot at all times. If the line is to be hot in winter and cold in summer, the loop should be spread half this amount so that it will be subject to but very slight bending stress either when hot or cold. Fig. 11 shows how an efficient expansion loop can be fabricated by welding through the use of short radius elbows.

Procedure Control

Successful results in the installation of a welded piping system can be insured only by the application of some definite regulations or methods that may be prescribed for the purpose of bringing together all of the essential elements for good workmanship. These embrace such elements as selecting the necessary materials of good welding quality, correctly designing the joints, properly preparing the parts for welding, predetermining the ability and knowledge of the welding operators, applying the most approved welding technique, and finally, properly testing the completed joints. This *procedure control* will not only insure safe welded construction, but also will make it possible to duplicate results at any place and at any time. Experience covering a period of many years has definitely proved the reliability of this method of application of welding, and has demonstrated that engineers and architects can use welded piping installations for all pressures and services with complete confidence of successful results if a well-established procedure control is followed.

Notes on Welding

Arc welding and oxy-acetylene welding are used with equal success in piping work.

Whenever a branch is joined to a main, templates should be used for cutting the hole through the wall of the main pipe and for preparing the end of the branch pipe. The branch pipe should not project inside the main but should fit snugly against the main so that with the process of welding no surplus metal will drop into the interior of the main. Any droppings should of course be removed.

When a branch is taken off from a main line and there is very little room for expansion it is good practice to weld in a reenforcing brace between the branch and the main line. This is particularly true on large piping where high temperatures occur.

In changing the direction of the run of pipe it is better to use bends than to use fittings and nipples. Special welding-fittings of uniform thickness are manufactured in all pipe sizes. These bends can be cut with the welding torch to any desired angle.

Since welding is a comparatively new industry there is some danger that failures may occur unless great care is taken with the workmanship. The following rules are basic in welding practice and should be followed rigidly:

1. A skilled operator with the torch or arc should be employed.
2. The proper welding rod and electrode must be supplied.
3. The choice of oxygen-acetylene or electric arc will be governed by the adaptability of each for the particular installation.
4. Templates shall be used for all cuts, with joints which are to be welded fitted closely prior to starting the welding.
5. Standard welding tools and equipment of recognized make should be used.

Capacity of Return Risers for Steam and Vapor Heating Systems

By **F. C. Houghten**[1] (*MEMBER*) and **Carl Gutberlet**[2] (*NON-MEMBER*)
Pittsburgh, Pa.

THIS paper marks the completion of the general investigation of the capacity of pipe for carrying steam, condensate, and air in various parts of steam heating systems, as outlined by the Technical Advisory Committee on Pipe Sizes for Heating Systems, and as carried on by the Research Laboratory in cooperation with the *Carnegie Institute of Technology*. Publication of the results of various phases of the study have appeared in the TRANSACTIONS[3] of the Society since 1922, and have formed the basis for the pipe size tables contained in THE GUIDE. This report deals with the capacity of return risers. The investigation was carried on in conjunction with the study of return mains, a report[4] of which was published recently.

Test Set-Up and Operation

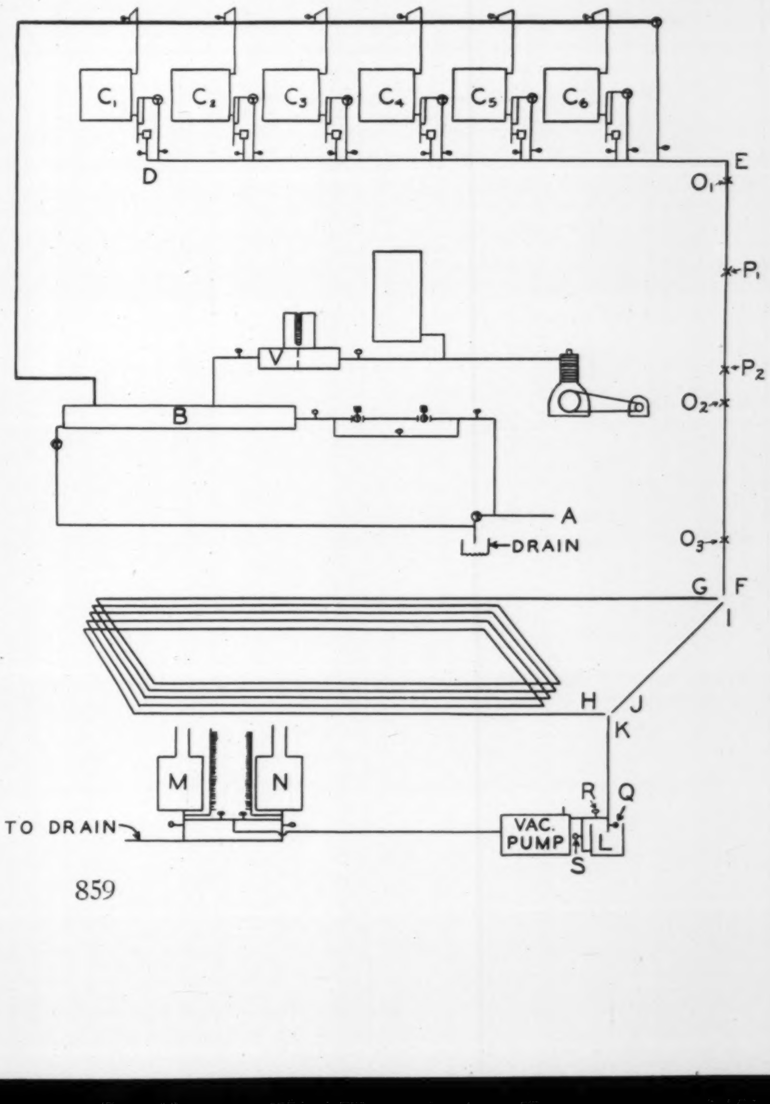
The test set-up, Fig. 1, varied only in minor details from that used in the study of return mains.[4] Steam from a high pressure line *A* was supplied to an 8 in. dripped main *B*, by means of suitable pressure reducing valves and hand-controlled throttling valves. From the main *B*, the steam was delivered to six unit heaters C_1 to C_6 varying in size from 150 to 2,200 equivalent sq ft of direct cast iron radiation. Any or all of the units could be turned on. The largest unit was provided with fan speed control. With this arrangement condensation could be supplied at any rate desired through the return main *DE* to the head of the riser, *E*. In order to give complete control over the rate of air carried by the return riser, provision was made for admitting air through the orifice flow meter *V* to the supply main *B*.

The base of the riser *F* could be connected to the receiving bucket *L* and the vacuum pump through 5 loops containing 540 ft of 1 in. dry return main, by closing the connection through *F, G, H,* and *K*. By closing the connection through *F, I, J,* and *K*, the return main could be eliminated. With either connection the system could be operated by gravity or by vacuum pump return. By opening valve *Q* and closing valve *R*, the system returned by gravity to the receiving bucket at atmospheric pressure, from which the condensation was pumped through *S* into the measuring tanks *M* or *N*. By closing *Q* and *S* and opening *R*, the system operated as a vacuum pump return, the pump delivering the condensate directly to the measuring tanks.

One-inch and ¾ in. risers were studied. Provision was made for observing the pressure at two points P_1 and P_2 along the riser in addition to points O_1, O_2, and O_3, by inserting the connection and manometer shown in Fig. 2. The large pipe connection prevented water from sealing across the pipe by capillary attraction, thus giving an erroneous reading. At each of the three points O_1, O_2, and O_3, a 28 sq ft radiator was drained into the riser. Fig. 3 shows the connection used for these radiators which was used primarily to indicate whether or not the riser was overloaded, but which also served to give three additional points at which pressures in the riser could be observed. When a radiator was not in use it was shut off from the return riser by closing the gate valve between the trap and the riser.

In operating the system, the rate of condensation and

FIG. 1—LABORATORY SET UP FOR STUDYING CAPACITIES OF RETURN RISERS



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[1] Director, Research Laboratory, A. S. H. V. E.

[2] Research Engineer, Research Laboratory, A. S. H. V. E.

[3] A. S. H. V. E. TRANSACTIONS AND JOURNAL, 1922 to date.

[4] *Capacity of Dry Return Mains for Steam and Vapor Heating Systems*—by F. C. Houghten and Carl Gutberlet, A. S. H. V. E. JOURNAL Section (Heating, Piping and Air Conditioning), August 1930.

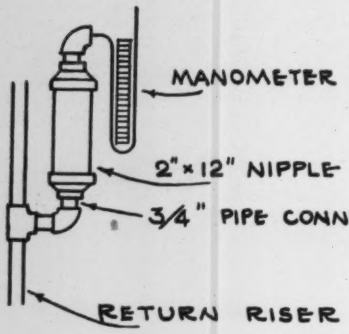


Fig. 2—(Left) Connection Used for Measuring Pressure in Return Risers

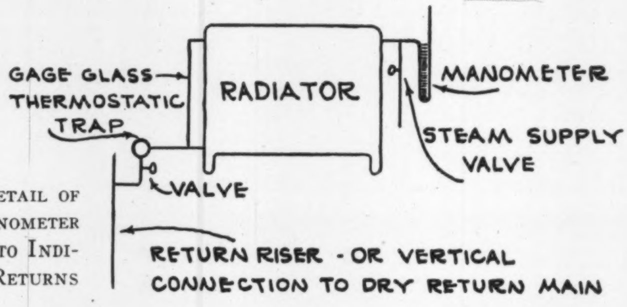


Fig. 3—(Right) Detail of Radiator and Manometer Connections Used to Indicate Overloaded Returns

air supply to the return riser was adjusted and maintained constant for a sufficient length of time to insure uniform operation and uniform temperature throughout the system. When all conditions were constant, the rate of condensation and the rate of air flow through the riser were determined. The condensate was measured by a graduated gage glass on the measuring tanks, and the air, by an orifice flow meter in the air supply line. The pressures at the points P_1 and P_2, and also those in the radiators, after the valves between the radiators and the riser were opened, were observed. If the pressure in the radiator before turning on the steam had been equal to or below atmospheric pressure, the steam supply valve was opened sufficiently to give a pressure of 1 oz. in the radiator as indicated by the manometer.

If the pressure in the radiator before turning on the steam was above atmospheric pressure, then the steam supply valve was opened sufficiently to give a manometer reading 1 oz. above this value. The time for completely heating the radiator was observed, and if more than 20 minutes were required the riser was considered to be overloaded and the test was not used.

Tests were made at condensation rates of 250, 500, 900, 1200, 1500, and 2000 lb per hour on the 1-in. riser and rates of 150, 300, 500, 900, and 1200 lb per hour on the ¾-in. risers. For each load studied the pressure drop along the riser was determined for various rates of carrying air up to the maximum which could be handled without overloading the riser, producing noise or requiring an excessive pressure drop.

Experimental Results

The pressures along the riser for several rates of handling condensation and air are plotted in Fig. 4. The pressure drop throughout the length of the riser in ounces per 100 ft length of run was calculated from curves similar to these in order to arrive at the pressure drop, condensation and air flow relationships desired.

The pressure drops in the riser for the various conditions studied as calculated from curves similar to those in Fig. 4 are plotted in Figs. 5 and 6 against cubic feet of air per hour handled by the 1-in. and ¾-in. risers, respectively.

It is of interest to note that for comparatively high rates of handling condensation with correspondingly low rates of handling air, negative pressure drops are observed, that is, with an increasing distance along the riser in the direction of flow (or from the top towards the bottom of the riser) the pressure increased rather than decreased. This is as would be expected upon careful consideration, for under these conditions acceleration of the descending water in the riser produces an aspirating effect tending to draw the air downward so as to produce a partial vacuum at the top of the riser.

For comparison, similar curves, *A* and *B*, Fig. 5, are shown giving the relation between pressure drop and cubic feet of air handled for 500 and 1500 lb condensation loads on a 1 in. horizontal dry return main pitched 1 in. in 16 ft. These curves are taken from an earlier laboratory report[4] of a study of dry return mains. It will be observed that the pressure drop for handling air and condensate at any rate is very much lower for the return riser than it is for the dry return main.

The pressure drop in ounces per 100 ft length of run along the 1-in. and ¾-in. risers is plotted in Figs. 7 and 8, respectively, against condensation load in pounds per hour, and in equivalent square feet of radiation for various rates of handling air ranging up to 150 cu ft per hour. It is again of interest to note the negative pressure drop found for certain rates of handling condensation and air. For the highest loads given for the ¾-in. riser, the *no air* curve tends to flatten out with increased load. This tendency would, no doubt, also be found for the *no air* curve on the 1-in. riser if the curve were carried out to higher loads. These curves must necessarily

Fig. 4—Pressure Drop Through the Return Risers for Various Air and Condensation Loads

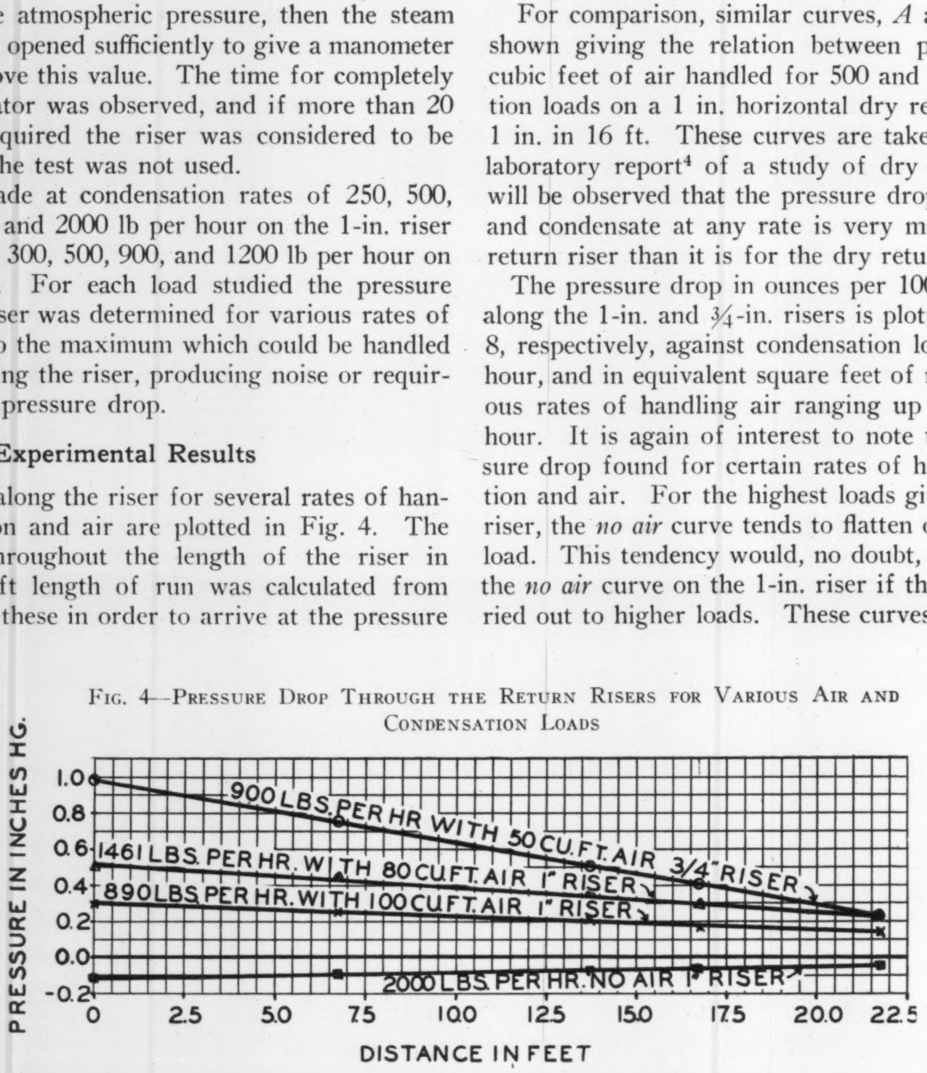
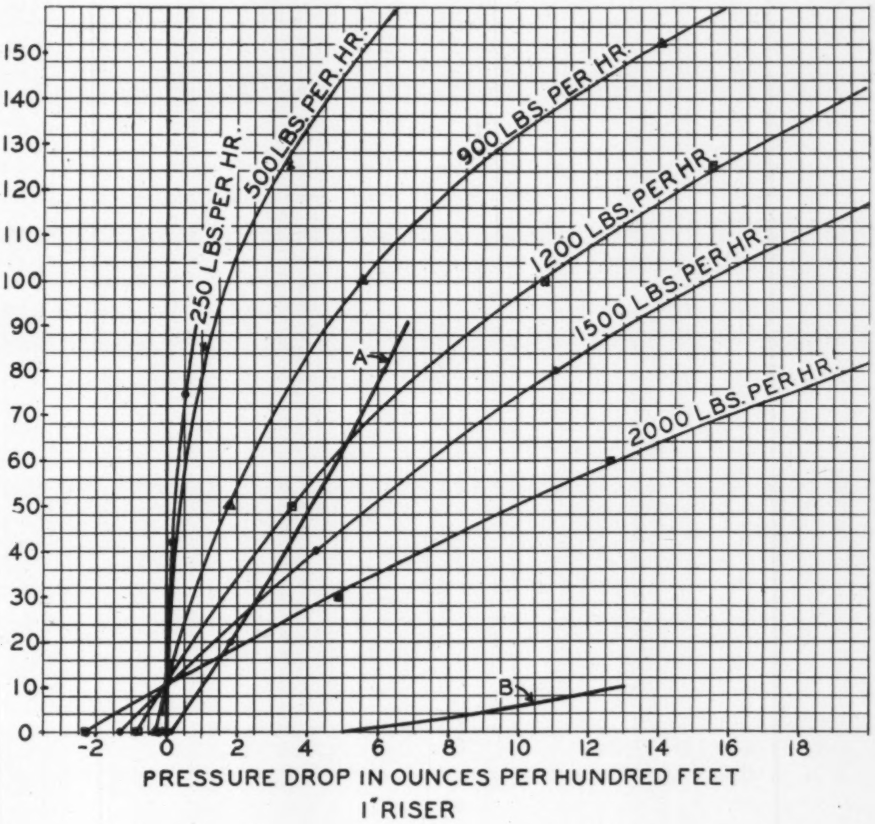


Fig. 5—Relation Between Pressure Drop in a 1-In. Return Riser and Air Carried for Various Condensation Rates



again curve upward and intersect the *zero pressure drop* line for some higher load just as the 5 cu ft curve for the ¾-in. riser after descending below the *zero pressure drop* line again ascends and recrosses the *zero* axis. The particular load at which the *no air* curve would again intersect the *zero pressure drop* line may be calculated for a vertical pipe filled with and carrying water alone at maximum capacity for gravity flow by applying the Darcy formula:

$$Q = ACF\sqrt{RS},$$

in which the various factors have the following significance and values for the 1-in. pipe:

Q = pounds of water per hour flowing through the pipe
A = area of the pipe
R = the mean hydraulic radius
S = Sine of the slope of the pipe = 1
C = a constant depending on the pipe
F = constant to change cubic foot of water per second to pounds per hour = 216,000

Kent[5] gives $AC\sqrt{R} = 0.06334$ for a 1-in. pipe and 0.02855 for a ¾-in. pipe. Therefore:

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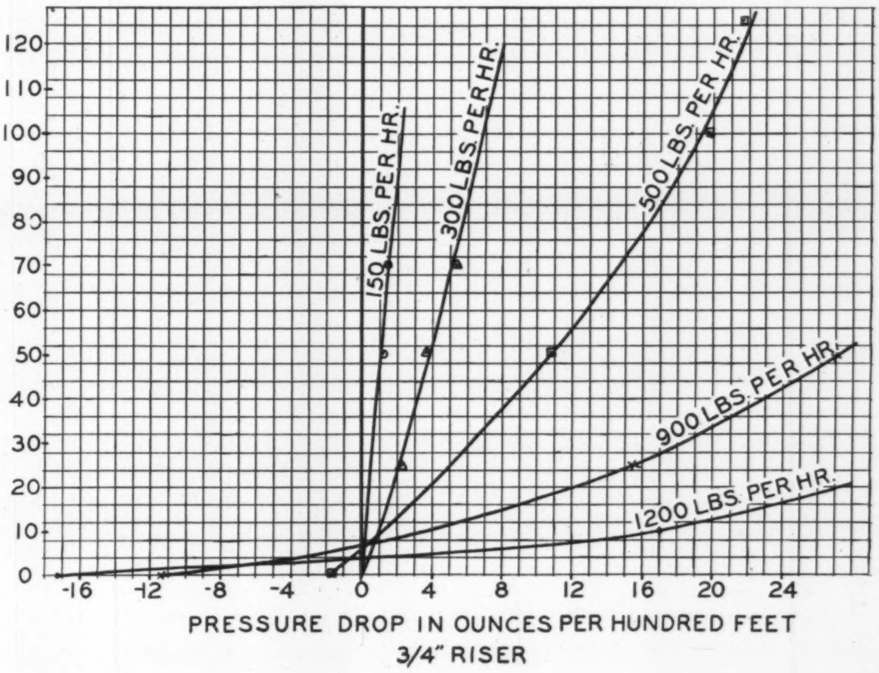
[5] Kent's Handbook, 1923, p. 754.

Fig. 6—Relation Between Pressure Drop in a ¾-In. Return Riser and Air Carried for Various Condensation Rates



$Q = 0.06334 \times 216{,}000 \times 1 = 13{,}682$ lb per hour for a 1-in. pipe

$Q = 0.02855 \times 216{,}000 \times 1 = 6{,}167$ lb per hour for a ¾-in. pipe

For the 1-in. vertical pipe filled with and carrying water alone at full capacity, due to gravity, the *no air* curve should again intersect the *zero pressure drop* line at a load of 13,682 lb per hour of water. Likewise the *no air* curve for the ¾-in. pipe should intersect the *zero pressure drop* line at a load of 6,167 lb of water per hour.

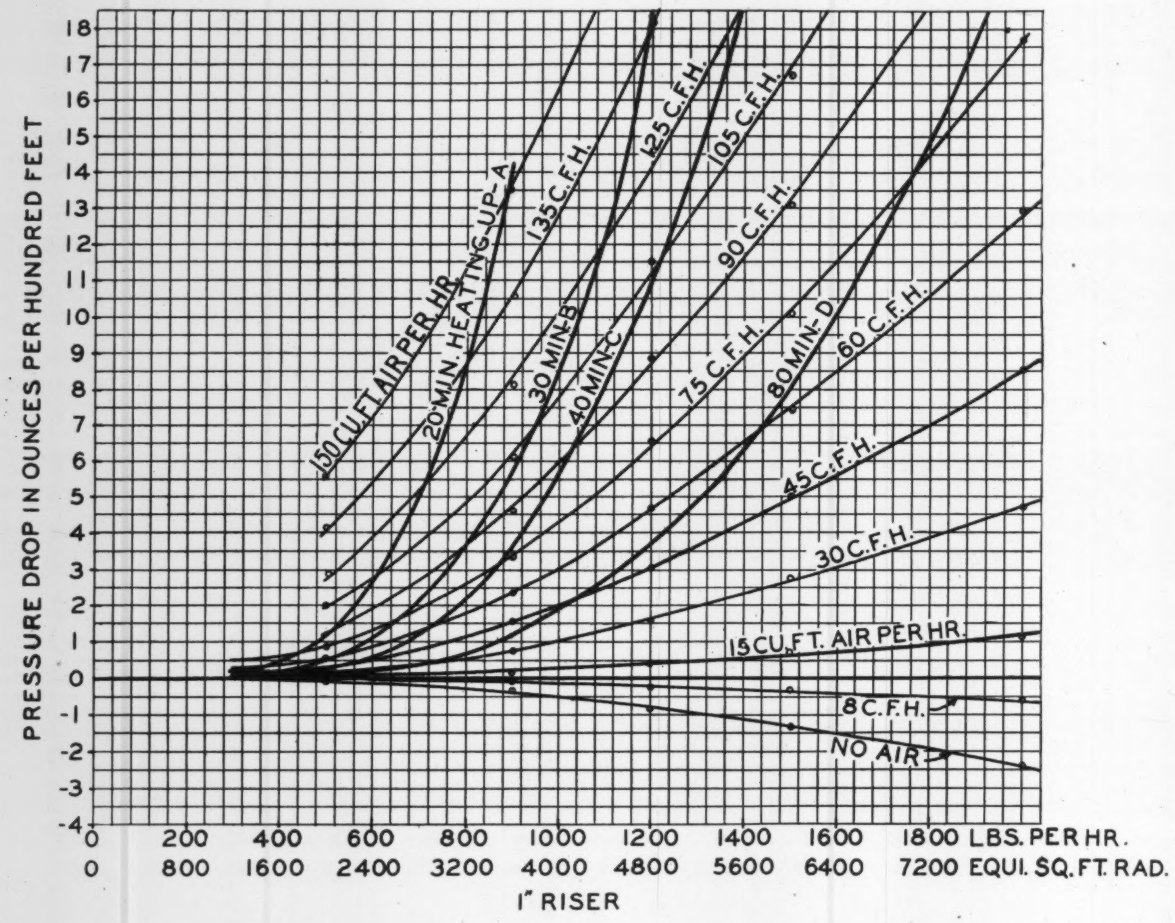


Fig. 7—Relation Between Pressure Drop and Condensation Load for Various Rates of Air Elimination Through a 1-In. Return Riser

It should be observed that the air loads indicated for the various curves in Figs. 4 to 8 represent the rates at which air was added to the steam. There was always in addition to this amount of air some non-condensable gases and probably also some water vapor so that the *no air* curve as plotted in Figs. 7 and 8 are really curves for some minimum amount of air, gas, and non-condensed vapors, and these curves, no doubt, actually intersect the *zero pressure drop* line at loads considerably below the theoretical value given by the Darcy formula.

In order to observe the quantity of air, non-condensable gases and vapors carried in the condensate, besides the quantity of air added to the system, a glass tube of the same internal diameter was placed in the bottom of the return risers and the system operated for a short time under different conditions. For condensation loads up to about 1200 lb per hour for the 1-in. riser, the descending water rarely filled the pipe, but descended in torrents down the sides and in a rain down the center. As the quantity of air handled increased, the descending water and air became more agitated. When no air was added, the flow was similar but with comparatively little agitation. For condensation loads above approximately 1200 lb per hour, slugs of water appeared, but even when no air was added these slugs were always interspersed with slugs of air and water spray.

The curves showing the relation of pressure drop to load for various rates of air handled as plotted in Figs. 7 and 8 give the data necessary for choosing the size of pipe required for any desired pressure drop and for any rate of air and condensation handled within the limits studied. However, before these curves may be used in practice it is necessary to arrive at the rate at which a riser in a given system will be required to handle air. This will depend upon a great many factors, including the volumetric capacity of the radiation and piping used in a given system and the rate at which it is desired to eliminate the enclosed air in heating up.

After all radiation in the system is heated up and the air eliminated, the rate at which air will be handled will be extremely low as shown by the tendency of the curves, The capacity for carrying condensation with a given pressure drop under such conditions is extremely high. It is obvious that the size of pipe required for any return riser must be based not upon its capacity to carry condensation when the system is entirely heated up or when little air or non-condensable vapors are being eliminated, but it must rather be based upon the rate of air elimination during the heating-up period, which will depend in a large measure upon the type of radiation and rate of heating up.

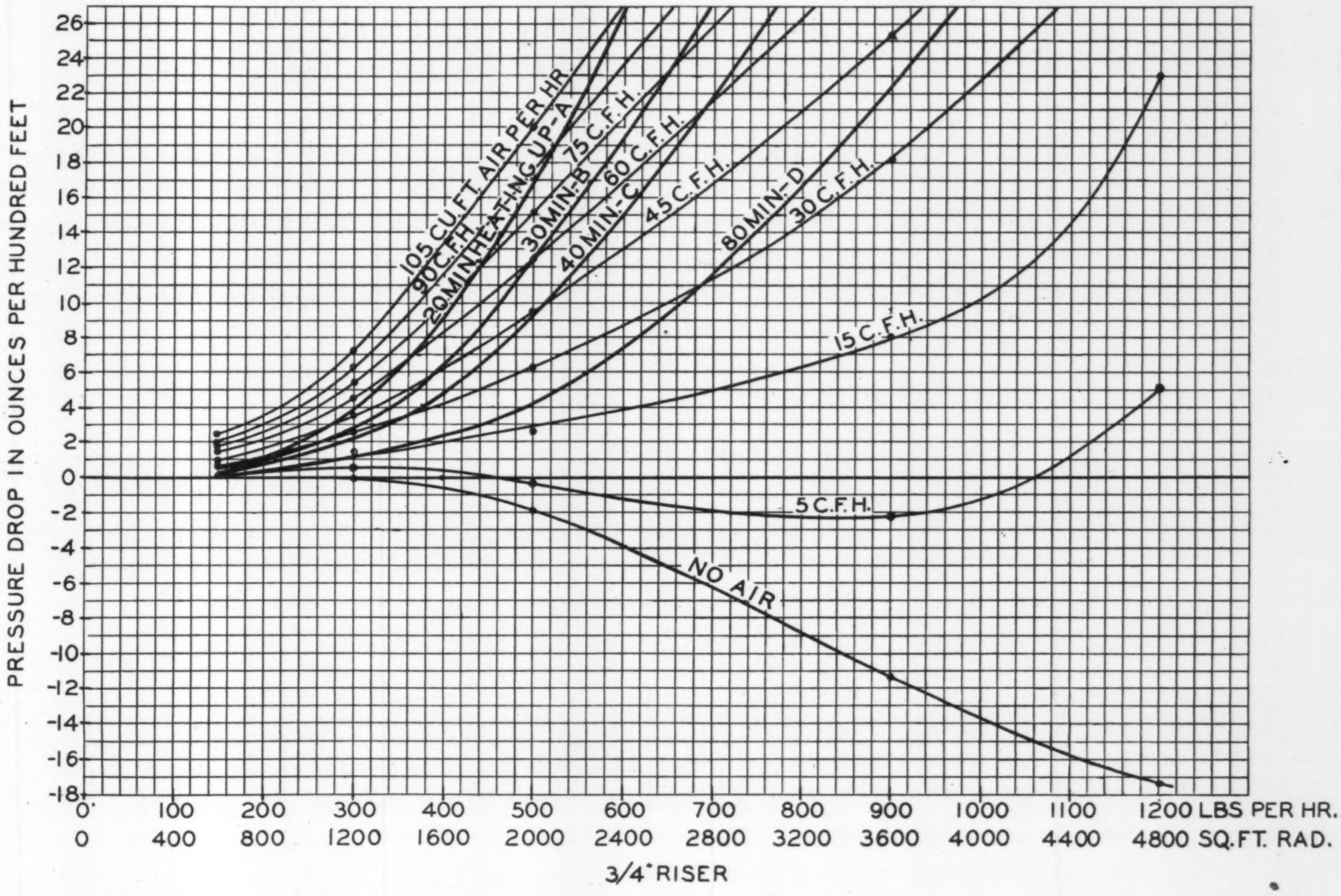


FIG. 8—RELATION BETWEEN PRESSURE DROP AND CONDENSATION LOAD FOR VARIOUS RATES OF AIR ELIMINATION THROUGH A ¾-IN. RETURN RISER

In an earlier laboratory report,[4] results were given of the average volumetric capacity found for a large number of sizes and makes of tubular cast iron radiators, which average was given as 0.0142 cu ft of air per square foot of radiation.

If it is assumed that the maximum rate of air elimination from a system while the return piping is handling both air and condensation at the maximum rate may be based upon the time of heating up the entire system and the elimination of a volume of air equal to the capacity of all the radiation served, it may then be possible to arrive at pressure drop, condensation relationships for various rates of heating up. Curves *A* to *D*, Figs. 7 and 8, give the pressure drop condensation relationships for rates of heating up ranging from 20 to 80 min. for systems in which all steam is condensed in tubular cast iron radiation having an average volumetric capacity of 0.0142 cu ft per square foot of radiation.

If air is eliminated in 20 minutes from a system in which all steam is condensed in tubular cast iron radiators having a volumetric capacity of 0.0142 cu ft per rated square foot, then the system must handle air at the rate of $3 \times 0.0142 = 0.0426$ cu ft of air per hour per square foot of radiation or 0.1704 cu ft of air per pound of condensation handled. The rates of handling air per pound of condensation given in curves *B*, *C*, and *D* were calculated similarly.

In Fig. 9 the limits in rates of handling air and condensation for satisfactory operation are given by curves *E* and *F* for the 1-in. and ¾-in. risers respectively. For determination of these limitations the riser was said to operate satisfactorily when the radiators O_1, O_2, and O_3 (Fig. 1) returning into it become entirely heated in 12 min without noise.

In all of the tests resulting in the data used in Figs. 4 to 9 the system was operated either by gravity or vacuum pump return as required to give satisfactory operation. If the system operated satisfactorily by gravity return it was also found to operate satisfactorily with vacuum pump return and with substantially the same pressure drop load relationships. When the vacuum pump was applied to a condition under which the system would operate satisfactorily by gravity the radiators heated up more quickly and, provided the vacuum pulled

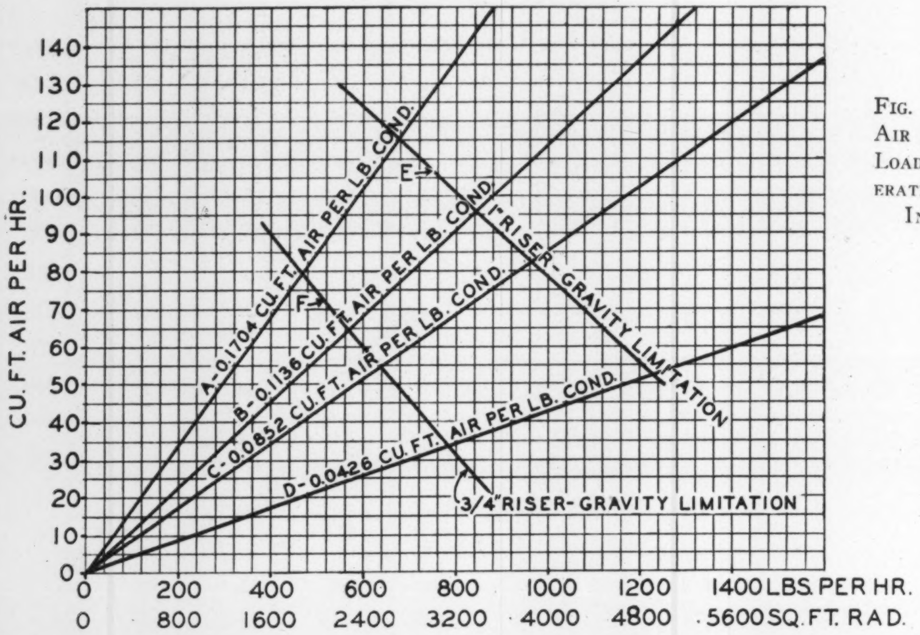


Fig. 9—Limitation of Air and Condensation Loads for Gravity Operation of 1-In. and ¾-In. Return Risers

by the pump was not great enough to produce an excessive vacuum (not over 1 in. of water) at the top of the riser, the pressure drop relationship for the same load condition was found to remain the same or to increase by a practically immeasurable amount. However, when the vacuum at the pump was increased so as to produce a greater vacuum at the top of the riser, the pressure drop for the same load condition increased. This increase as pointed out in an earlier laboratory report on dry return mains was probably the result of reevaporation of condensate upon passing through the trap when a large pressure drop through it existed.

Most of the tests resulting in the data presented were made with direct connection between the base of the riser and the receiving bucket or the vacuum pump. Especially for gravity return, this connection was made large so that there would be substantially no pressure drop between the base of the riser and the receiving bucket. In other words, the pressure at the base of the riser was substantially atmospheric pressure. Some tests were made, however, on all conditions reported in which the condensation from the riser returned through the 540 ft dry return main. Excepting for very small loads, the insertion of the return main between the base of the riser and the receiving bucket gave unsatisfactory operation of the riser when operating by gravity. However, if the vacuum pump was applied so as to give atmospheric pressure at the base of the riser *F*, the same pressure drop load relationships were found for the riser.

Table 1—Capacity of 1 in. Return Riser in Square Feet of Equivalent Radiation for Various Rates of Air Elimination

Experimental Results

Pressure Drop Ounces per 100 Ft	Period of Air Elimination			
	20 Min	30 Min	40 Min	80 Min
¼	1360	1640	1940	2560
½	1620	2000	2300	2960
1	1936	2336	2704	3472
2	2240	2760	3136	4032
3	2472	3040	3440	4512
4	2648	3272	3680	4912

Table 2—Capacity of 1 in. Return Riser in Square Feet of Equivalent Radiation for Various Rates of Air Elimination

Experimental Results Less 20 Per Cent

Pressure Drop Ounces Per 100 Ft	Period of Air Elimination				
	20 Min	30 Min	40 Min	80 Min	From Guide 1930[a]
¼	1088	1312	1552	2048	
½	1296	1600	1840	2368	
1	1549	1869	2163	2778	1200
2	1792	2208	2509	3226	1704
3	1978	2432	2752	3610	
4	2118	2618	2944	3930	2400

[a] Capacities of return risers for vacuum installations. No rate of air elimination given.

Application of the Data in Sizing Return Risers

The relationships given in the curves, Figs. 4 to 9, may be used for developing tables giving capacities of 1 in. and ¾ in. return risers. The acceptance of such tabular values must, however, be based upon certain assumptions, including the allowable pressure drop, the rate of heating up or air elimination and any factor of safety desired for covering other contingent factors. These assumptions cannot be determined by laboratory test but must be based upon experience gained through practice and judgment.

Table 1 gives the capacity of a 1-in. return riser in equivalent square feet of radiation for various pressure drops and various rates of heating up a system in which all steam is condensed in tubular cast iron radiators having a volumetric capacity of 0.0142 cu ft per square foot of radiation.

Table 2 gives capacities 20 per cent lower than those given for the same conditions in Table 1. This 20 per cent reduction in capacity may be considered a factor of safety for variation in internal diameter and internal smoothness of pipe, constrictions due to improper cutting, reaming, and the use of dope, and other contingent factors. The acceptance of a factor of 20 per cent is in

TABLE 3—CAPACITY OF ¾ IN. RETURN RISER IN SQUARE FEET OF EQUIVALENT RADIATION FOR VARIOUS RATES OF AIR ELIMINATION

Experimental Results

PRESSURE DROP OUNCES PER 100 FT	PERIOD OF AIR ELIMINATION			
	20 MIN	30 MIN	40 MIN	80 MIN
½	600	680	720	860
1	764	820	892	1136
2	964	1056	1152	1496
3	1108	1228	1360	1776
4	1208	1368	1500	2116

TABLE 4—CAPACITY OF ¾ IN. RETURN RISER IN SQUARE FEET OF EQUIVALENT RADIATION FOR VARIOUS RATES OF AIR ELIMINATION

Experimental Results Less 20 Per Cent

PRESSURE DROP OUNCES PER 100 FT	PERIOD OF AIR ELIMINATION				
	20 MIN	30 MIN	40 MIN	80 MIN	FROM GUIDE 1930
½	480	544	576	688	
1	611	656	714	909	700
2	771	845	922	1197	994
3	886	982	1088	1421	
4	966	1094	1120	1693	1400

conformity with the earlier practice of the committee which developed the pipe size tables for the supply side of steam heating systems contained in THE GUIDE, 1930, and is not the result of this laboratory investigation. Tables 3 and 4 give similar data to Tables 1 and 2 for a ¾-in. return riser.

Table 5 is based upon Fig. 9 and gives minimum time in which various amounts of tubular cast iron radiation can be heated up when returning through a 1-in. or a ¾-in. riser, and also the accompanying maximum rate of air elimination. These data are given for both the 1-in. and ¾-in. risers. Table 6 is derived from Table 5 by the application of a 20 per cent factor of safety.

Tables 2 and 4 give, for the sake of comparison, capacities for return risers for various pressure drops as used in the pipe size tables in THE GUIDE, 1930. No required time for heating up is given in connection with tables in THE GUIDE. It is of interest to note that for a 1-ounce pressure drop, which is usually considered best practice, the values in THE GUIDE are considerably below the values resulting from this investigation and the application of a 20 per cent factor of safety for a 20-minute heating-up period. However, for higher pressure drops this difference disappears and finally reverses so that for a 4-in. pressure drop THE GUIDE values are higher, although not enough so that the 20 per cent factor of safety would allow the riser to operate satisfactorily in the absence of any contingent factors tending to decrease its capacity. In the case of the ¾-in. riser (Fig. 4) all GUIDE values show higher capacities in comparison to those resulting from this investigation, and the application of the 20 per cent factor of safety for a 20-min. heating-up period. For a 1-oz. pressure drop, the values in THE GUIDE correspond to a heating-up period of approximately 40 min., whereas for a 4-oz pressure drop the values in THE GUIDE correspond to a longer heating-up period.

It is not the purpose of this investigation to establish the desirable heating-up period on which to base pipe size capacities. However, it may be of interest to point out that a 20-min. heating-up period is probably much shorter than desirable for most buildings for the reason that in most instances it is not necessary or desirable to heat up all radiation from the cold condition in this period of time.

Summary and Conclusions

1. As a result of this investigation, curves in Figs. 7 and 8 give the relation between condensation carried by the return riser, pressure drop through the riser, and air carried by it on which pipe size capacities may be based.

2. Capacities of return risers are given in tabular form based upon experimental values found in the laboratory and also upon the application of a factor of safety of 20 per cent to these values.

TABLE 5—MAXIMUM RATE OF AIR ELIMINATION FROM TUBULAR CAST IRON RADIATION FOR VARIOUS CONDENSATION LOADS AND CORRESPONDING HEATING UP PERIOD FOR 1 IN. AND ¾ IN. GRAVITY RETURN RISERS

Experimental Results

CAPACITY SQ FT EQUIVALENT RADIATION	MAXIMUM RATE OF AIR ELIMINATION			
	1 INCH RISER		¾ INCH RISER	
	TIME, MIN(a)	CU FT AIR/HR	TIME, MIN(a)	CU FT AIR/HR
1600			15	89.5
2000			23	74.5
2400	16	124	34	60.0
2800	20	112.5	50	45.5
3200	27	100.5	88	31.0
3600	34	89.5		
4000	43	78.5		
4400	56	67.0		
4800	74	55.5		

[a] Time necessary for eliminating air from a system in which all steam is condensed in tubular cast iron radiation having a volumetric capacity of 0.0142 cu ft per equivalent square foot of radiation.

TABLE 6—MAXIMUM RATE OF AIR ELIMINATION FROM TUBULAR CAST IRON RADIATION FOR VARIOUS CONDENSATION LOADS AND CORRESPONDING HEATING UP PERIOD FOR 1 IN. AND ¾ IN. GRAVITY RETURN RISERS

Experimental Results Less 20 Per Cent

CAPACITY SQ FT EQUIVALENT RADIATION	MAXIMUM RATE OF AIR ELIMINATION			
	1 INCH RISER		¾ INCH RISER	
	TIME, MIN(a)	CU FT AIR/HR	TIME, MIN(a)	CU FT AIR/HR
1280			15	89.5
1600			23	74.5
1920	16	124	34	60.0
2240	20	112.5	50	45.5
2560	27	100.5	88	31.0
2880	34	89.5		
3200	43	78.5		
3520	56	67.0		
3840	74	55.5		

[a] Time necessary for eliminating air from a system in which all steam is condensed in tubular cast iron radiation having a volumetric capacity of 0.0142 cu ft per equivalent square foot of radiation.

Development of the Ventilation System of the Holland Tunnel

By A. C. Davis[1], New York
MEMBER

THE Holland Tunnel between New York City and Jersey City, N. J., was the first ventilated vehicular tunnel started in the United States. Prior to starting the construction of this tunnel, the principal vehicular tunnels in operation were the Blackwall Tunnel under the Thames River, in London, England, completed in 1897; the Rotherhithe Tunnel, also under the Thames, London, completed in 1908; the Glasgow Harbor Tunnel, Glasgow, Scotland, completed in 1895; and the Elbe Tunnel, Hamburg, Germany, completed in 1910. None of these tunnels at the time of completion was equipped with mechanical ventilation. In 1925 a mechanical ventilating system was installed in the Rotherhithe Tunnel.

With no precedent as a guide, the study of the ventilation of The Holland Tunnel was begun in 1919. The following problems had to be solved:

1. Amount and composition of exhaust gases from automobile engines while operating on the highways.
2. Effect of such gases on animal life.
3. Dilution necessary in order that persons may breathe the exhaust gases for short periods of time without injurious effects.
4. System of ventilation best suited to vehicular tunnels.
5. Power required to ventilate the tunnel.
6. Demonstration of the operation of the system.

The best available data on the gases given off by automobile engines were those of Herbert Chase.[2] The tests on which these data were based were conducted on the streets of New York, on varying grades, types of pavement and at various speeds. Owing to the limited number of types of automobiles tested and to the dissimilarity of grades and speeds as compared with those proposed for the tunnel, definite conclusions could not be based on these studies. For similar reasons, the results obtained by Alleman, Ballantyne, Watson or Hood and Kudlich could not be used.

Exhaust Gases from Automobile Engines

An agreement was entered into with the U. S. Bureau of Mines to perform the tests on the streets of Pittsburgh, Pa., having grades and pavement similar to those proposed for the tunnel. The selection of types of motor vehicles (see Table 1) for tests was based on traffic observations at the ferries across the Hudson River. All vehicles tested were those in actual use on the streets of Pittsburgh; no adjustments were made in carburetors and no change was made in the brand of gasoline used.

In general, the tests were made under the following conditions:

1. Car standing, engine racing.
2. Car standing, engine idling.
3. Car accelerating from rest to 15 mph up 3 per cent grade.
4. Car running 3, 10 and 15 mph, up 3 per cent grade.
5. Car running 3, 10 and 15 mph down 3 per cent grade.
6. Car accelerating from rest to 15 mph on level grade.
7. Car running 3, 10 and 15 mph on level grade.

TABLE 1—CLASSES OF MOTOR VEHICLES TESTED

CLASS	DESCRIPTION	NO. OF CARS TESTED
1	Five-passenger cars	19
2	Seven-passenger cars	13
3	Trucks up to 1½ tons carrying capacity	12
4	Trucks 1½ to 3 tons, carrying capacity	22
5	Trucks 3½ to 4 tons, carrying capacity	17
6	Trucks 5 tons and over, carrying capacity	18
	Total	101

Fig. 1 shows the apparatus for collecting the exhaust gas sample from the exhaust manifold of an automobile. The samples were collected in sampling tubes and taken to the chemical laboratory for analysis by the modified Orsat method. Fig. 2 shows the testing apparatus installed on a truck. In the front is shown the apparatus for collecting the exhaust gas sample, and in the rear, the gasoline measuring apparatus is shown.

Summary of Important Results

From the standpoint of tunnel ventilation, the results of the road tests on 101 motor vehicles were briefly as follows:

1. Automobile exhaust gas consists of carbon dioxide (CO_2), carbon monoxide (CO), hydrogen (H_2), methane (CH_4), oxygen (O_2), nitrogen (N_2) and water vapor. The relative proportions of these constituents varies greatly in the exhaust from different motors, depending upon carburetor adjustment, degree of atomization, compression and other variable factors.

2. The important constituent of exhaust gas, as regards tunnel ventilation, is carbon monoxide (CO).

3. The CO found in the exhaust gas of the various industrial cars varied from 0.5 to 14.0 per cent.

4. The higher percentages of CO are produced when the throttle is nearly closed, as when running down grade or the car is standing and engine idling. The largest quantity of CO is produced when the gasoline consumption is greatest, as with cars accelerating or running upgrade at maximum speed.

From these tests and with the proposed number and types of vehicles using the tunnel at the specified speeds, it was calculated that the following quantities of CO would be produced in the tunnel:

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[1] Assistant Superintendent, The Holland Tunnel.

[2] See *The Automobile*, by Herbert Chase (Vol. 30).

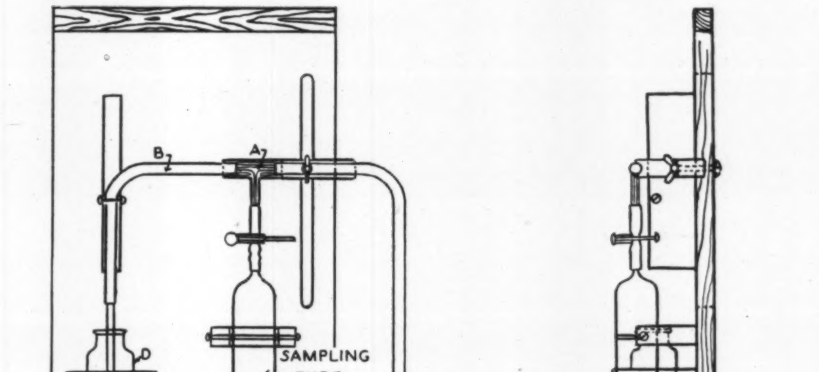


Fig. 1—Apparatus for Collecting Exhaust Gas Sample from the Exhaust Manifold of an Automobile

On the up grades—0.118 cfm per foot of tunnel.
On the down grades—0.05 cfm per foot of tunnel.
On the level grades—0.09 cfm per foot of tunnel.

The motor vehicles to use the tunnel were classified as follows: Passenger cars, trucks under 2 tons capacity, 2 to 5 tons, and 5 tons and over. The ventilation requirements were based on 20 per cent passenger cars and 80 per cent trucks.

Effect of Automobile Exhaust Gases on Animal Life

To determine the effect of automobile exhaust gases on animal life and to ascertain the dilution necessary to make such gases safe to breathe, investigations were carried on at the U. S. Bureau of Mines Experimental Station at *Yale University*. The experiments were performed in two types of chambers, one a gastight chamber having a capacity of 226 cu ft (Fig. 3), and the other a brick chamber having a capacity of 12,000 cu ft (Fig. 4).

Members of the investigating staff spent periods of one hour in the gastight chamber containing chemically prepared carbon monoxide varying from 2 to 10 parts per 10,000 parts of air. Groups of students, as well as members of the investigating staff, spent periods of one hour in the larger chamber in various concentrations of carbon monoxide, produced by the automobile shown in Fig. 4. The rear wheels of the automobile were removed and replaced by paddle-wheels to distribute the CO throughout the chamber. Fig. 4 also shows a meter for measuring the quantity of exhaust gas produced by the engine. Samples of the air in the chamber were taken periodically to determine the concentration. Subjects were examined by a doctor before and after each exposure, and samples of blood were taken and analyzed for CO content.

The conclusions drawn from these tests were, briefly, as follows:

1. Exhaust gas of automobiles contains no substance which is toxic to any appreciable extent, other than carbon monoxide.
2. Gasoline engines, with cylinders missing or when cold, may throw off disagreeable vapors irritating to the eyes and nauseating to some persons.
3. The physiological effects of carbon monoxide are wholly due to the union of this gas with the hemoglobin. To whatever extent the hemoglobin is so combined, by that amount it is rendered incapable of transporting oxygen to the organs and tissues of the body. The combination of CO and hemoglobin is a reversible reaction, so that when a person returns to fresh air the CO is gradually eliminated.
4. Of all physical signs and tests of CO poisoning,

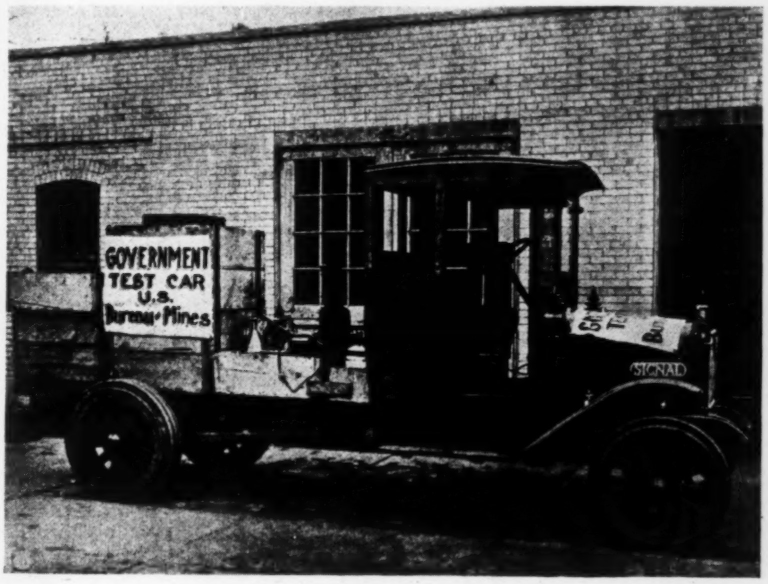


Fig. 2—Exhaust Gas Testing Apparatus Installed on a Truck

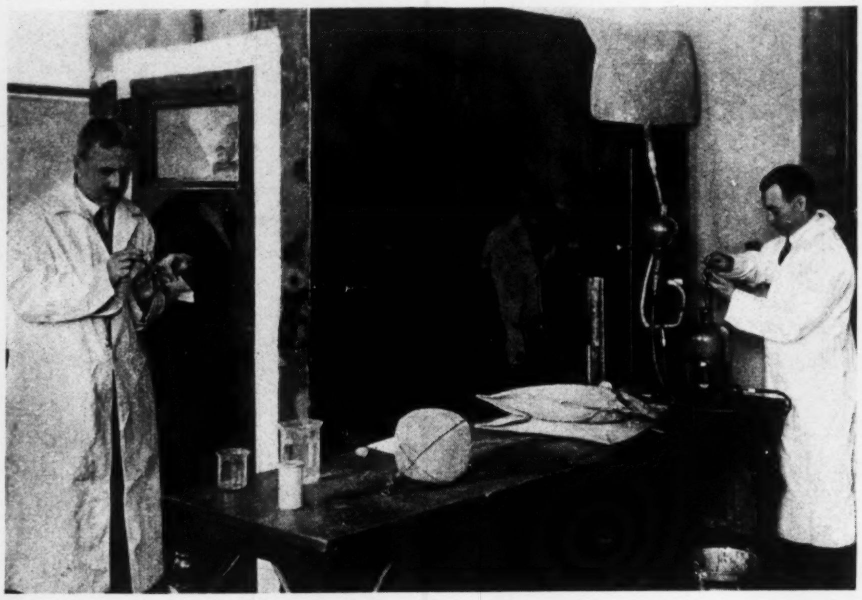


FIG. 3—GAS-TIGHT CHAMBER FOR DETERMINING EFFECT OF AUTOMOBILE EXHAUST GASES ON ANIMAL LIFE

headache proved the most definite and reliable. Concentration of gas too weak or periods of exposure too short to induce this sign, are to be considered harmless.

Dilution of Exhaust Gases Necessary

No one had this symptom (headache) to an appreciable degree after a period of one hour in the chamber with 4 parts of CO in 10,000 parts of air. With 6 parts the degree of effect, if any, was usually very slight, while with 8 parts there was decided discomfort for some hours, although not enough to interfere with the continuance of efficient work in the laboratory or at the desk. In any atmosphere containing a uniform amount of CO, almost all the gas inhaled at first is absorbed, but the successive increments of gas absorbed thereafter grow less and less as time goes on. If a man exercises sufficiently to double his volume of breathing, he absorbs as much CO in one-half hour as he does at rest in one hour.

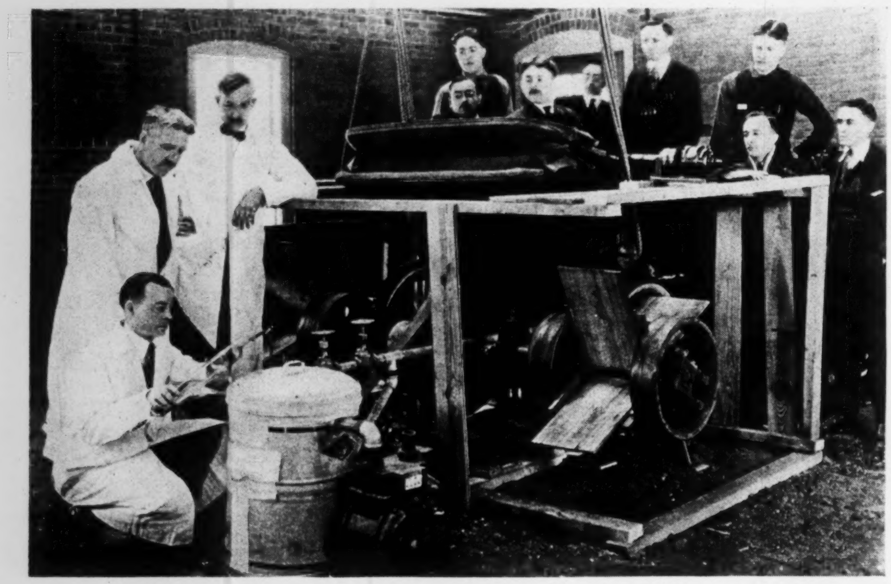


FIG. 4—BRICK CHAMBER FOR DETERMINING EFFECT OF AUTOMOBILE EXHAUST GASES ON ANIMAL LIFE

Fig. 5 shows the rates of saturation of carbon monoxide hemoglobin and percentages of frontal headaches at various times of exposure when subjects are at rest, as when riding in an automobile. Based on the results of the investigations and on the estimated traffic capacity with assumed classification of traffic, it was determined that the following quantities of fresh air would be required per minute per foot of tunnel:

Down grade—125 cu ft
Level grade—225 cu ft
Up grade—295 cu ft

On this basis a total of 3,761,000 cu ft of fresh air per minute was required for the entire tunnel. The amount of air to be exhausted was calculated to be 4 per cent in excess of the fresh air supply.

System of Ventilation Best Adapted

Prior to starting these investigations, studies were made to determine the system of ventilation best adapted to vehicular tunnels. These studies, which involved 21

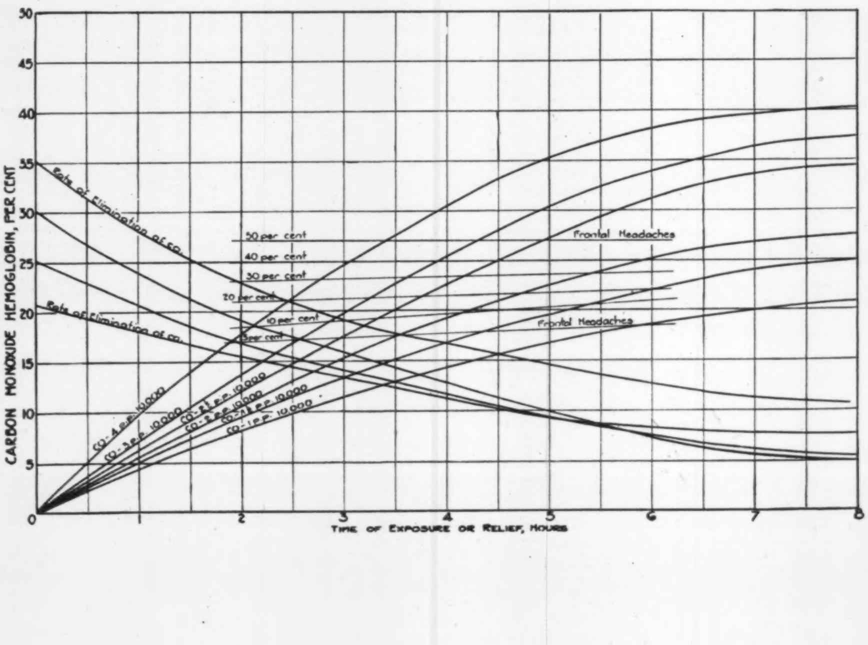


FIG. 5—RATES OF SATURATION AND ELIMINATION OF CO—HEMOGLOBIN AND PERCENTAGES OF FRONTAL HEADACHES AT VARIOUS TIMES OF EXPOSURE OR RELIEF, SUBJECTS AT REST

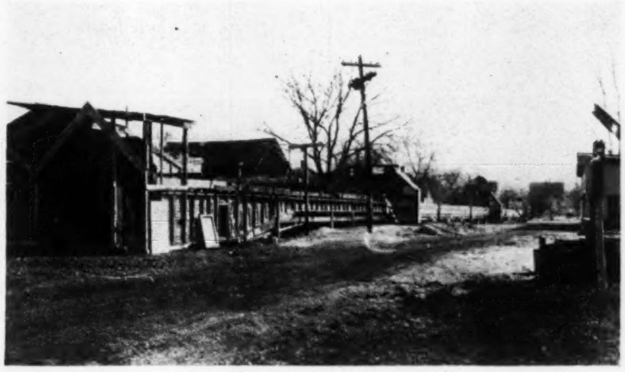


FIG. 6—GENERAL VIEW OF LARGE CONCRETE DUCT TESTED

schemes, included longitudinal, transverse and combinations of these two methods of ventilation. As a result of these experiments it was decided to use the transverse method of ventilation, but no definite decision was made as to whether the upper or lower duct should be used for the fresh air supply. The studies indicated that the lower duct should be used for the fresh air supply and on this basis experiments were made to determine the power required to ventilate the tunnel.

Power Required to Ventilate the Tunnel

These experiments were conducted at the Engineering Experiment Station of the *University of Illinois*, under an agreement with the U. S. Bureau of Mines. The scope of the work, in part, was as follows:

1. Determination of the coefficient of friction for the flow of air in concrete ducts, as proposed for the tunnel.
2. Determination of the formulae for calculating the pressure necessary to deliver the air through concrete ducts of constant cross-section, with the air leaving or entering in uniform quantities at uniform intervals.
3. Determination of pressures necessary to convey air through various types of concrete elbows, louvres and in fresh air and exhaust air ports.

The experiments under items (1) and (2) were performed in a concrete model, the linear dimensions of which were one-half of those of the proposed duct under the roadway, and on a one-tenth scale metal and wood duct model. Fig. 6 is a general view of the large concrete duct. It was 300 ft long, with a wood duct 75 ft long connecting it to the test fan which had a capacity of 105,000 cfm and was driven by a 300 hp motor.

Outlets to half-scale dimensions of those proposed for the tunnel were provided on each side of the duct at uniform intervals throughout the 300 ft length. These outlets were provided with adjustable shutters for controlling the flow of air. The interior of the duct was smooth and free from fins. Three air measurement stations were provided, and the end of the duct farthest from the fan was equipped with adjustable gates for the control of pressures. Fig. 7 is a general view of small duct which was one-fifth the dimension of the concrete duct; its length was 60 ft and it was connected by means of a metal duct 29 ft long to a fan direct connected to a 10 hp d-c motor. The small duct was equipped with outlets similar to those of the large duct, and it also had three air measurement stations. Tests were run in the large duct to determine the coefficients of friction with various velocities of air flow, the inside of outlet ports being closed off flush with the inner duct surface.

A large series of experiments were conducted to determine the pressure required to expel varying quantities of air from the outlets for various duct lengths. In these tests the outlets had to be adjusted so that uniform quantities of air were expelled from each outlet. As a result of this study, a formula was developed for calculating the pressure required for supplying air to a duct from which the air is removed in equal quantities at uniform intervals along its length. After these tests were completed the fan inlets and outlets of the supply fan were reversed so that the air was drawn through the duct after entering the side ports, thus representing the exhaust air duct of the tunnel. These tests showed that much greater power was required to draw the air through the duct (corresponding to exhaust duct) than to force it through (corresponding to fresh air duct).

To minimize the power requirements for exhausting the air, studies were made to design a port which would be practicable to install in the ceiling slab of the tunnel and allow the air to enter the duct as nearly parallel to the main flow as possible, instead of at right angles. The small model duct was designed with a removable top. One top was solid, and with it on the duct all tests made on the large duct were duplicated. From these tests and a knowledge of the law of dimensional homogeneity, a formula was developed for the calculation of the pressure requirements to force the air through the duct.

After completion of the blower duct tests the solid top of the small duct was removed and replaced with another top in which various types of exhaust ports were investigated. From these experiments it was found that the most satisfactory port was one with an inclination of 45 deg in the direction of the main flow in the duct. With this type of port a series of tests was made from

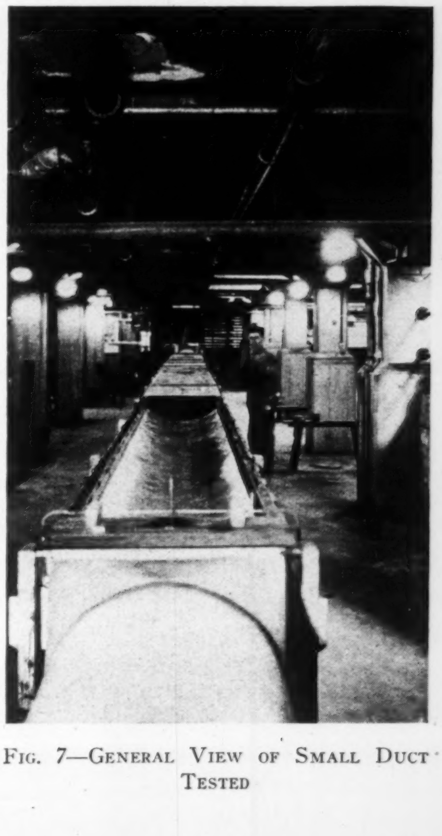


FIG. 7—GENERAL VIEW OF SMALL DUCT TESTED

which a formula for the exhaust duct pressures was developed. From these tests the following formulae were developed:

Coefficient of friction:

$$f = a + \frac{b}{m^2 w^2} \qquad (1)$$

where

$a = 0.0035$
$b = 0.01433$
$m =$ area of duct in square feet divided by perimeter in feet
$w =$ velocity of air in feet per second $= \frac{Q}{A}$
$Q =$ cubic feet of air per minute passing the end of the duct nearest the fan
$A =$ area of duct in square feet

For blowing condition:

$$\text{T. P.} = \frac{12y}{D}\left\{\frac{W^2}{2g}\left[\frac{aLZ^3}{3m} - \frac{1}{2}(1-K)Z^2\right] + \frac{bLZ}{2gm^3}\right\} + P_o \qquad (2)$$

where

T. P. = Total pressure in inches of water
$y =$ Density of air in duct in pounds per cubic foot
$D =$ Density of water in pounds per cubic foot
$W_1 =$ Velocity of air entering the duct in feet per second
$g = 32.2$ feet per second, per second
$a = 0.0035$
$b = 0.01433$
$L =$ Total length of duct in feet
$Z = \frac{L - X}{L}$
$X =$ distance from entrance of duct to any chosen section
$m =$ Hydraulic mean radius in feet = area of duct divided by perimeter
$K = 0.615$, a factor allowing for turbulence in the duct
$p_o =$ Bulkhead pressure in inches of water

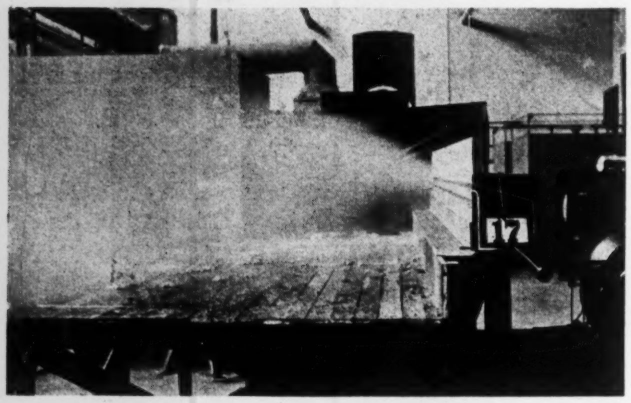


FIG. 8—FRESH AIR PORTS WITH EXPANSION CHAMBER

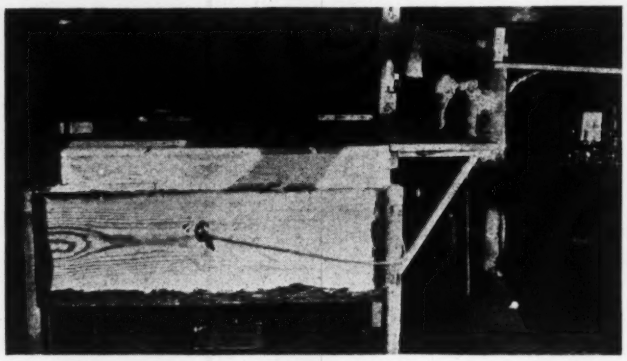


FIG. 9—FULL SIZE EXHAUST AIR PORTS

For exhaust condition:

$$S.P. = \frac{12y}{D}\left\{\frac{W_1^2}{2g}\left[\frac{aL}{(3+c)m}Z^3 + \frac{3}{2+c}Z^3\right] + \frac{bL}{2gm^3(1+c)}Z\right\} + p_0 \qquad (3)$$

where

S. P. = the suction head or negative static pressure
$c = 0.25$ for flow in excess of 200 cu ft per minute per foot of tunnel. For lower rates, $c = 0.20$

All other terms have the same significance as in equation (2).

The exhaust equation holds for all lengths of ducts, but that for blowing requires certain other considerations for duct lengths wherein the minimum static pressure falls outside of the main duct from which air is being taken off. The minimum static pressure in a duct must be sufficient to handle the maximum quantity of air to pass the port. To determine what these minimum pressures were for various quantities of air, tests were made on full size fresh air ports with expansion chamber, as shown in Fig. 8, and also on full size exhaust air ports, as shown in Fig. 9. The results of these tests were plotted and used in the setting of the ports in the tunnel.

In addition to the foregoing tests, other tests were made for the determination of pressure required for the passage of air through elbows of various sizes and shapes, and through louvres. After having calculated the pressure required to handle the air through an entire duct section, the air horsepower was calculated by using the formula:

$$\text{hp} = \frac{P \times Q \times 62.4}{33000 \times 12} \qquad (4)$$

where

hp = Horsepower
$P =$ Pressure in inches of water
$Q =$ Total quantity of air handled in cu ft per minute

The total horsepower to be delivered by the motor was determined by applying the efficiencies of the fan, transmission and motor.

Demonstration of the Operation of the System

After the completion of the investigations already described, it was necessary to demonstrate the operation of the proposed system on a large scale and also to demonstrate whether the air supply to the driveway should be from the duct over the driveway or the duct under it. This demonstration was conducted in the U. S. Bureau of Mines experimental mine at Bruceton, Pa., part of which was reconstructed for this purpose, providing a miniature tunnel, oval in plan, with a driveway length of 400 ft, located about 1,200 ft in from the mountainside and entirely shut off from the outside atmosphere except through a drift connecting the tunnel to the ventilation plant located outside. Fig. 10 shows a plan of the layout. The tunnel was 9 ft wide and 8 ft high, with a roadway sufficient for one line of automobiles. Ventilation ducts were constructed above the ceiling and under the roadway. Provision was made so that the flow of air to and from the tunnel could be reversed, using alternately the upper and lower ducts for the fresh air supply.

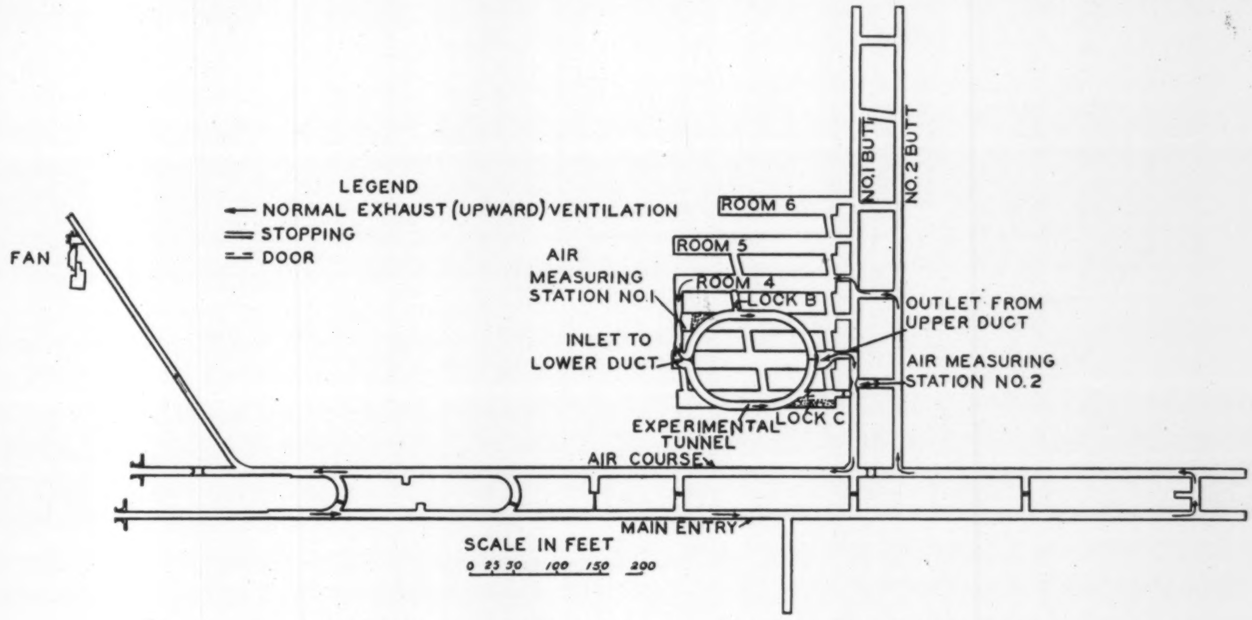


Fig. 10—Layout of U. S. Bureau of Mines Experiment Mine

Tests were made in the experimental tunnel with as many as eight automobiles at a time, operating as they would in the proposed tunnel. Some tests were made with the fresh air supply from the duct under the roadway into the driveway just above the roadway, and exhausted through ports in the ceiling to the duct above. Others were made with the air supply from the duct above the ceiling into the driveway and exhausted through ports just above the roadway into the lower duct. Samples of air were taken from various points in the driveway and also from the moving automobiles, and analyzed.

The results of the tests showed that the two methods of transverse air movement were practicable for tunnel ventilation, and that the best method, from the standpoint of economy and safety against fire hazard, was the one in which the fresh air is introduced from the duct under the roadway and exhausted through the duct above the ceiling. With this method the CO passes the breathing zone of drivers once, whereas, with the reverse method, the CO passes twice, once on its way up from the exhaust pipe, and again on its way down due to the action of downward air currents from the upper duct.

The amount of CO given off by the automobile during these tests checked very closely the results previously obtained from the investigation of the amount and composition of exhaust gases from automobiles. The physiological effects of CO were again carefully checked during these tests; the results agreed closely with the findings of the earlier investigations.

Tests were also made to determine the effect of hot weather on the absorption of CO by humans. For this purpose the tunnel air was heated by gas jets to a dry-bulb temperature of 103 F and a wet-bulb temperature of 95 F. It was found that heat up to this degree had no effect on the rate of absorption of CO. Observations were also made on the flow of heat through the concrete walls of the experimental tunnel under conditions approximating those proposed for The Holland Tunnel.

The original investigation to determine the physiological effects of CO on passengers using the tunnel did not include a determination of the effect of long exposures to CO in concentrations up to maximum allowance, such as would be experienced by the employes on duty in the tunnel. To obtain this information, additional research work was done under an agreement with the Bureau of Mines at the Bureau's Experimental Station at Pittsburgh, Pa. The information obtained was used in the setting up of working schedules for the tunnel employes. The investigation was carried out in a gas-tight chamber especially equipped for the purpose.

Six subjects were simultaneously exposed in each test. Of these, three remained at rest and three exercised mildly. The exercise consisted of stepping up on a bench 24 in. high a number of times in rapid succession each half hour. The subjects were exposed to concentrations of CO ranging from pure air to 4 parts in 10,000 parts of air, for periods of from four to seven hours each day for 68 days in succession.

As in the previous investigations, the frontal headache was the first symptom and generally occurred when about 20 per cent CO hemoglobin saturation is attained. Fig. 5 shows the relation of the number and time of occurrence of frontal headache to percentage of CO saturation of blood, and also the rate of elimination of CO. These curves make possible the determination of the proper length of shifts for employees working in the tunnel under various percentages of CO in the air.

The investigation also showed definitely that the administration of oxygen after exposures to CO greatly diminished the time necessary for desaturation. In the operation of the tunnel the administration of oxygen has not been necessary as no symptoms of CO poisoning have developed.

The investigations show what can be done by the close co-operation between the engineer and the research scientist. The engineer analyzed the problem with which he was confronted, reduced it to its several component parts, and stated the different points on which research work was required. The scientist conducted the research work in such a way as to furnish the information on the various phases necessary for the solution of the problem as a whole.

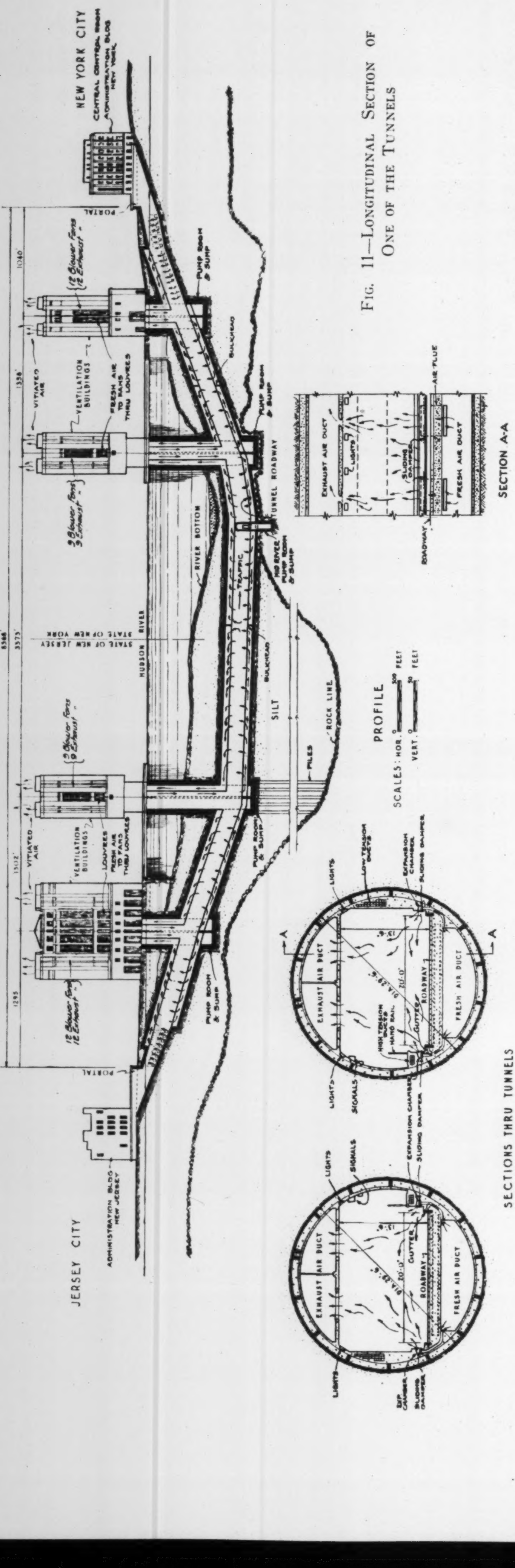

NEW YORK CITY
CENTRAL CONTROL ROOM ADMINISTRATION BLDG NEW YORK
PORTAL
12 Blower Fans 12 Exhaust
VENTILATION BUILDINGS
VITIATED AIR
FRESH AIR TO FANS THRU LOUVRES
PUMP ROOM & SUMP
BULKHEAD
TUNNEL ROADWAY
RIVER BOTTOM
TRAFFIC
9 Blower Fans 9 Exhaust
HUDSON RIVER
STATE OF NEW YORK
STATE OF NEW JERSEY
SILT
ROCK LINE
PILES
PROFILE
SCALES: HOR. VERT.
1040'
1358'
8368'
3575'
1502'
1295'
JERSEY CITY
ADMINISTRATION BLDG NEW JERSEY
SECTION A-A
EXHAUST AIR DUCT
FRESH AIR DUCT
AIR FLUE
ROADWAY
SLIDING DAMPER
EXPANSION CHAMBER
LIGHTS
SIGNALS
GUTTER
LOW TENSION DUCTS
HIGH TENSION DUCTS
HAND RAIL
SECTIONS THRU TUNNELS
SCALE
FEET

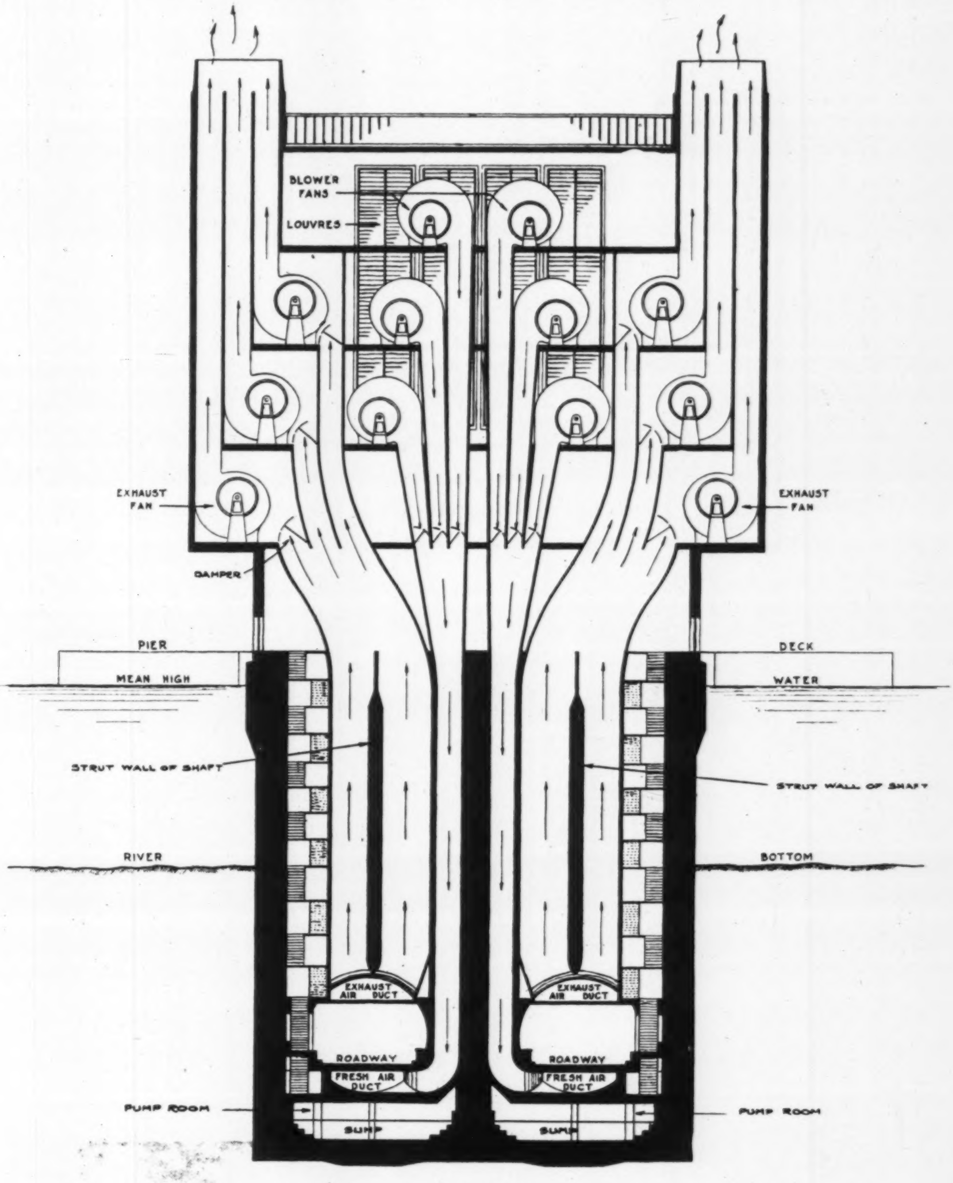
Fig. 11—Longitudinal Section of One of the Tunnels

The system of ventilation adopted for The Holland Tunnel is shown in Figs. 11 and 12. Fig. 11 shows a longitudinal section of one of the tunnels, with the four ventilation buildings. The fresh air duct is below the roadway, and from this duct air is taken off through flues located 10 to 15 ft apart in either side of duct. This air is delivered to the continuous expansion chamber on either side of the driveway. The face plate on these expansion chambers is adjustable and is set so that, when the maximum supply of air is delivered, the velocity of air through the slot is 1,000 ft per minute. In order to control the supply of air from the duct, slide dampers are provided in the expansion chambers on top of the flues. The vitiated or exhaust air duct is above the ceiling slab, and the air from the driveway enters this duct through exhaust ports located 10 to 15 ft apart. These exhaust ports are special in design, being made of cast iron ⅜ in. thick and provided with an adjustable zinc slide. Each tunnel has seven fresh air ducts and seven exhaust ducts, and each duct terminates in three openings, one to each fan. Thus, there are 21 fresh air fans and 21 exhaust air fans to each tunnel, or a total of 84 fans for the two tunnels. These fans are located as follows: 12 exhaust fans and 12 fresh air fans in each of the land ventilation buildings, and 9 exhaust fans and 9 fresh air fans in each river ventilation building.

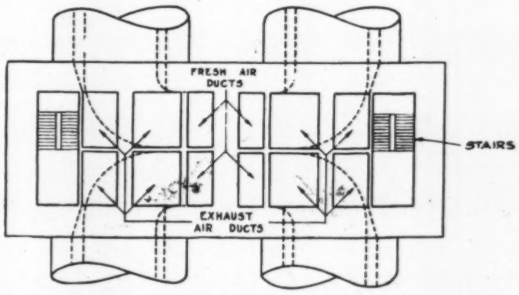
The three terminals of each duct are provided with dampers so that, when a fan is not in operation, the duct terminal may be shut off to prevent short-circuiting of air from other fans, as shown in Fig. 12. Air for the fresh air fans is drawn in through the louvres in the sides of the buildings, and the vitiated air from the tunnel is exhausted through the stacks at the corners of the buildings.

The ventilating fans are backward-curved blade type of fan, with a high static efficiency and a non-overloading horsepower characteristic. The housings are double width and double inlet, constructed of No. 10 gage steel, and are split so as to permit of ready dismantling for repairs.

Fan bearings are self-aligning type, and are made with a high grade of Babbitt metal. The thrust collars on fan shafts are machined on the shaft and bear against Babbitted surfaces in the bearings. Inspection doors are provided and, in the case of exhaust fans, the housings are connected to drains so that water entering the fan through the exhaust stacks may be drained off, there being no dampers in exhaust stacks to prevent entry of water or snow.


BLOWER FANS
LOUVRES
EXHAUST FAN
EXHAUST FAN
DAMPER
PIER
DECK
MEAN HIGH
WATER
STRUT WALL OF SHAFT
STRUT WALL OF SHAFT
RIVER
BOTTOM
EXHAUST AIR DUCT
EXHAUST AIR DUCT
ROADWAY
ROADWAY
FRESH AIR DUCT
FRESH AIR DUCT
PUMP ROOM
PUMP ROOM
SUMP
SUMP

SECTION
TYPICAL RIVER VENTILATION BLDG


FRESH AIR DUCTS
STAIRS
EXHAUST AIR DUCTS

PLAN

FIG. 12 — METHOD OF VENTILATION OF THE HOLLAND TUNNEL

The fan bearings are mounted on pedestals, which in turn are mounted on machined sole plates. Sole plates are anchored to steel structures of the buildings. The fan housings are anchored to the concrete floor. The fans are driven by slip ring motors through chain transmission. The chains run in fully enclosed oiltight cases.

Motors of 100 hp or less are 440-volt, 60-cycle, 3-phase; others are 2,200-volt, 60-cycle, 3-phase.

Each exhaust fan is in an airtight chamber along with its transmission, but the motor is outside of the room. These exhaust fan chambers are provided with remov-

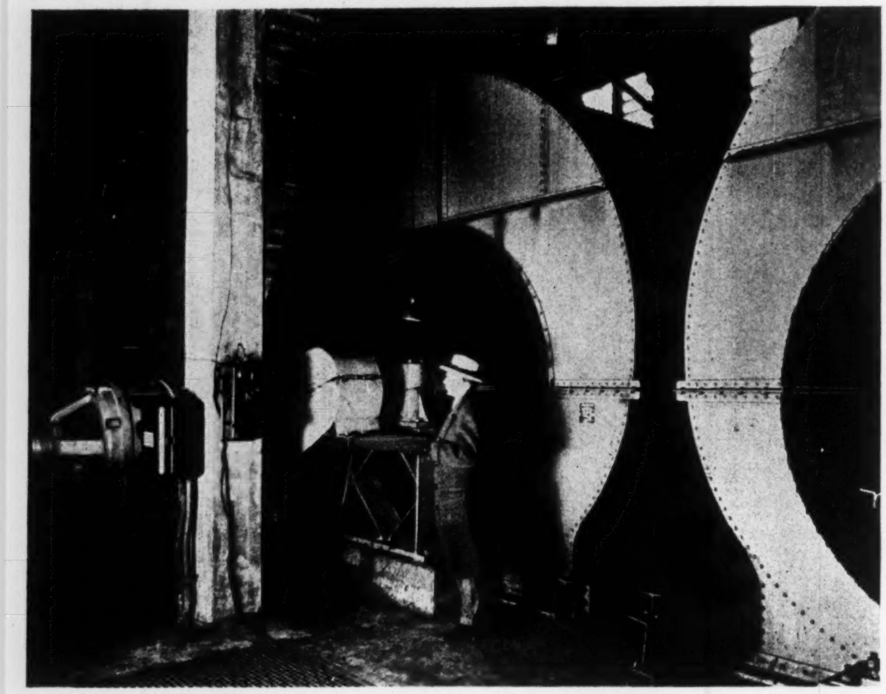


Fig. 13 — Typical Outside Air Fan

able steel partitions which were erected after the fan was installed, and can be readily removed in case of fan renewals. Exit and entrance from these chambers is by air locks.

Fig. 13 shows a typical fresh air fan, and Fig. 14 shows an exhaust fan installation. The fan motors can be operated either from the main control room in the New York Administration Building, from the local control board in the ventilation buildings, or from the local controls near each motor.

When the equipment was installed, the control was designed for two accelerating speeds and three running speeds. The three running speeds were approximately ½, ¾ and full speed. The control system is now being changed to include two more running speeds between ¾ and full speeds, to effect further economy in the operation of the system.

Characteristics of the Ventilation System

The characteristics of the ventilation system are as follows:

1. Uniform distribution of fresh air throughout the length of the tunnel.
2. The ventilation of the tunnel is not affected by outside atmospheric conditions.
3. Air supplied to or exhausted from the driveway can be definitely controlled.
4. The exhaust gases are quickly diluted and removed.
5. There are no high velocity air currents in the driveway due to the air supply.
6. Smoke from fires in the tunnel is quickly removed and prevented from spreading.

To operate the 84 fans requires 6,120 hp.

To handle the maximum quantity of air for normal ventilation requires 56 fans, which require 4,100 hp; 28 of these fans are fresh air fans which supply 3,761,000 cfm.

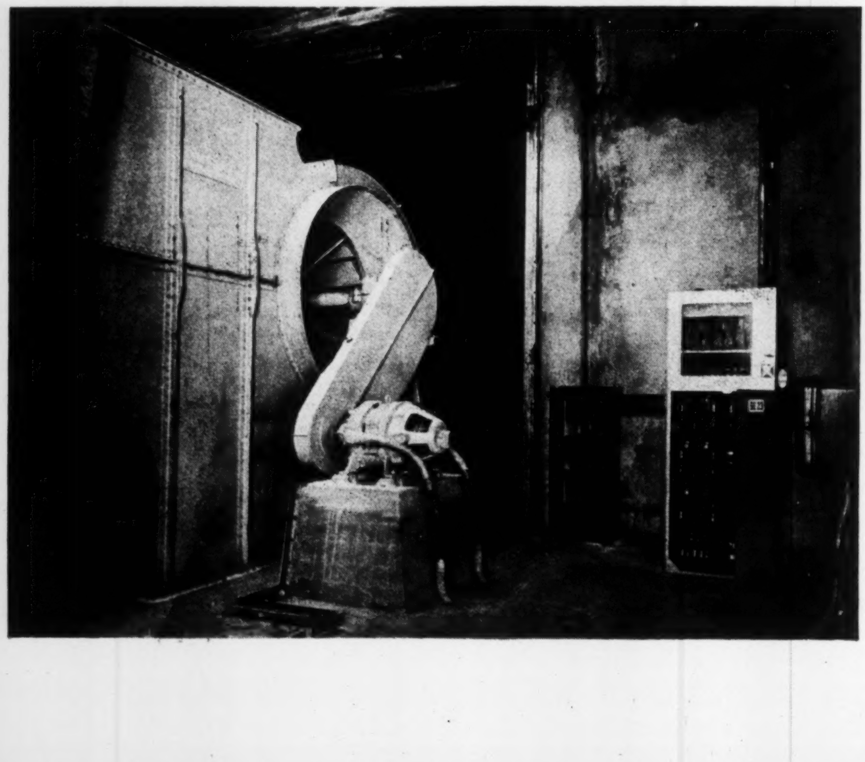


Fig. 14—Exhaust Fan Installation

Heat Transfer Through Air Spaces

Including a Resumé of Some of the Important Investigations

By Paul D. Close[1], New York
MEMBER

THE conductance of an air space depends upon several factors, among which are the dimensions, the character and shape of the surfaces enclosing it, and their position, that is, whether they are horizontal or vertical, the temperature difference between these surfaces, and the mean temperature of these surfaces.

Heat is transferred across an air space of any appreciable size by means of all three methods of heat transfer, namely, radiation, convection and conduction. Since the molecules of air are not as closely associated as those of solid materials, the motion of the molecules is less readily communicated by conduction through an air space than through most solids. Hence, if the heat transferred across an air space by radiation and convection could be eliminated, it would have a high degree of insulation. This, in effect, is what takes place when an air space is replaced by an insulation, that is, the radiation and convection elements are eliminated, and although the rate of heat transfer through the insulation by conduction may be greater than the rate of transfer through the air space by conduction, the total rate of heat transfer usually is reduced to a considerable extent.

Investigations of Various Experimenters

Notable among the experimenters who have investigated the transfer of heat through air spaces are the U. S. Bureau of Standards, the University of Minnesota, the Department of Scientific and Industrial Research of Great Britain, Messrs. H. Krueger and A. Eriksson (Swedish experimenters), Pennsylvania State College, Dr. Karl Hencky (Technical University at Munich), and Wilhelm Nusselt, a private lecturer of the University of Dresden.

The following is a brief summary of the results obtained by some of the experimenters:

U. S. Bureau of Standards: The results obtained at the Bureau of Standards on air spaces were published in Vol. 3 of the *A. S. R. E. Journal,* September 1916. A summary of the same work is in the *Smithsonian Physical Tables,* 7th Edition, page 253. (See also *The Thermal Conductivity of Heat Insulators,* by M. S. Van Dusen, A. S. H. V. E. Transactions, Vol. 26, 1920.)

Table 1, which is based on this report, indicates to some extent the effect of width upon the conductance of an air space. These experiments were made on metallic surfacs of low emissivity and then calculated over to the basis of air spaces enclosed between building materials. In making these calculations ordinary building materials were assigned an emissivity of 0.70 instead of the now commonly accepted 0.90. This makes the reported conductances lower than they should be. Also, if the results were recalculated on the basis of an emissivity of 0.90, it would tend to lessen the effect of width upon conductance.

The figures showing the rather large decrease in conductance with increased height of air space are not entirely reliable. Although there is, no doubt, some decrease of conductance with increased height, yet according to the results shown by other sources, this effect is slight.

If the air spaces 8 in. high are considered, it would appear from this table that an air space of ½ in. is about the optimum width. For spaces less than ½ in. the conductance increases rapidly with decreasing width. For spaces wider than ½ in. the decrease of conductance with increasing width is slight. Apparently, for air spaces with a low temperature gradient such as are usually found in practice, there is no width at which the conductance is at a well defined minimum.

The writer is informed that a more thorough investigation of this subject has been made by the U. S. Bureau of Standards recently and that the results will be published at an early date.

Special Report No. 9 of the Department of Scientific and Industrial Research. (*British.*) This report gives the results of some rather extended work on air spaces.

TABLE 1—CONDUCTANCE OF VERTICAL AIR SPACES BASED ON TESTS CONDUCTED AT U. S. BUREAU OF STANDARDS [a]

Bounding Surfaces Ordinary Building Paper

WIDTH OF SPACE INCHES	HEIGHT OF SPACE INCHES	TEMPERATURE DIFFERENCE —DEGREES FAHRENHEIT—			
		18	27	36	45
⅛	8	2.08	2.12	2.16	2.21
¼	8	1.33	1.38	1.38	1.42
⅜	8	1.08	1.13	1.13	1.17
½	8	0.96	1.00	1.00	1.04
⅝	8	0.92	0.96	1.00	1.04
¾	8	0.92	0.96	1.04	1.08
⅞	8	0.92	1.00	1.04	1.08
1	8	0.92	1.00	1.04	1.08
½	24	0.92	0.96	0.96	1.00
1	24	0.795	0.834	0.915	0.96
2	24	0.78	0.83	0.88	0.96
3	24	0.78	0.82	0.88	0.92

[a] Conductance expressed in Btu per hour per square foot per degree Fahrenheit.

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[1] Technical Secretary, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

No effect on the conductance of an air space was noted when the height of the air space was increased from 2 to 4 ft. It was found that the effective convection is confined to less than ½ in. for the report states that, "from 2 in. down to ½ in. the width of air space has no influence on the magnitude of the transfer by convection." The transfer by convection is only part (about 1/3 to ½) of the total, but the other part, radiation, is entirely independent of width.

University of Illinois, Engineering Experiment Station. Bulletin No. 102. In this bulletin the conductance of a ½ in. air space between window glass was given a value of 1.77 Btu per hour per square foot per degree Fahrenheit. A chart is given showing the effect of temperature gradient upon conductance. In plotting conductance against temperature difference between the enclosing walls the authors have given three points, these three points being obtained from experiments on widths varying from ½ in. to 4 in. Apparently, there is practically no variation of conductance with width within this range.

It is also stated that L. A. Harding, in a Pennsylvania State College Experiment Station Bulletin, reported a conductance of 1.66 for air spaces from 1 to 6 in. in width. Wilhelm Nusselt, who has written upon the subject of heat transfer, has stated that air spaces greater than ¾ in. give no additional value for heat insulating purposes.

Pennsylvania State College Bulletin No. 30. The following conclusions were based upon some postal box tests.

"Based upon 1.0 as the insulating value of three 1½-in. air spaces 5 in. high and 15½ in. long, three 1-in. air spaces equal 0.88 and three ½-in. air spaces equal 0.70. Hence, if three air spaces are to be used, the thickest of the three tested will be the best insulator on the basis of total transmission.

"Based on 1.0 as the insulating value (for total transmission) for three ½-in. air spaces, two ½-in. spaces equal 0.79 and one ½-in. space equals 0.59. Due to surface effects, it is evident that a 1-in. air space will not give twice the insulating value of a ½-in. air space."

These results, by themselves, should not be given too much credence because they were obtained by a test method which is now in general disuse because of unreliability.

Ingeniors Vetenskaps Akademien Handlingar No. 36 by H. Krueger and A. Eriksson (1924). Tests were made upon air spaces from 0.2 to 45 cm (0.079 to 17.7 in.) in width and enclosed between a variety of building materials. The resistance of the enclosed air space was calculated in most cases by subtracting the surface resistances (outside surface resistances) and the resistance of the enclosing material from the total air to air resistance of the structure.

TABLE 2—RESISTANCES OF AIR SPACES ENCLOSED BY GLASS FOR VARIOUS WIDTHS AND HEIGHTS BASED ON TESTS CONDUCTED BY H. KRUEGER AND A. ERIKSSON

WIDTH OF AIR SPACE —CENTIMETERS	HEIGHT OF AIR SPACE —CENTIMETERS	RESISTANCE OF AIR SPACE —RECIPROCAL OF CONDUCTANCE IN BTU UNITS
1.0	20	0.73
1.0	120	0.63
3.0	120	1.07
3.5	38	0.93
10.0	20	0.98
10.0	120	0.93
5.0	120	1.12

TABLE 3—RESISTANCES OF AIR SPACES FOR VARIOUS WIDTHS BASED ON CURVE PLOTTED FROM TESTS CONDUCTED BY H. KRUEGER AND A. ERIKSSON

WIDTH OF AIR SPACE —INCHES	RESISTANCE OF AIR SPACE —RECIPROCAL OF CONDUCTANCE IN BTU UNITS
¼	0.61
½	0.73
¾	0.79
1	0.84
1½	0.92
2	0.97
4	1.08

Table 2 contains part of the data given in this report. These figures show a slight effect of height upon resistance and also a tendency for a maximum resistance at a width of from three to five centimeters in width.

Taking all the results into consideration the authors say that it is noteworthy that height has but slight effect on the resistance of an air space. On page 129 they have plotted the results and drawn a curve through the rather scattering points. A few values taken from this curve are given in Table 3. They pertain to air spaces enclosed between any ordinary building material.

Die Wärmeverluste durch ebene Wände, by Dr. Karl Hencky of the Technical University at Munich. Dr. Hencky analyzes the phenomenon of air space heat transfer with all the precision that available data permit. In his analysis he breaks up the total heat transmitted through air spaces into the components (1) direct conduction from air particle to air particle of still air, (2) convection due to air currents, (3) radiation from one surface to the other. He lists a series of values for the heat flow through still air by conduction which varies with the temperature. For a temperature of about 68 F the heat transferred by conduction is about 0.16 Btu per hour per square foot per degree Fahrenheit.

The second component, namely, that due to convection, depends for its magnitude upon the thickness of the air space as well as upon the height of the air space. W. Nusselt conducted some tests to determine the convection factor and the figures quoted by Hencky are based upon Nusselt's work. Apparently, the only variable which entered into Nusselt's investigation was the thickness.

The third component, namely, that due to radiation, is based upon the well known Stefan-Boltzman law showing the proportionality of heat radiated to the fourth degree of the absolute temperature. The constant involved in the Nusselt equation founded upon the Stefan-Boltzman law, involves the radiation constants of the two opposing surfaces and the radiation constant of an absolutely black body. Dr. Hencky worked out an equation for the heat transmitted by radiation in terms of mean temperature on the centigrade scale, and the various radiation constants of the surface involved. He then combined the three equations dealing with the heat transmitted by conduction, convection and radiation into one equation in which the proportionality factor is analogous to the usual coefficient of conductivity figure for an ordinary insulating material. He calls this the equivalent conductivity of an air space. It is that conductivity of some insulating material which would occupy the same space as the air space and give the same results. There is this difference between the equivalent conductivity and the coefficient of conductivity of most ordinary insulating materials, that the equivalent conductivity is not a constant at all, whereas the coefficient of conductivity of most insulating materials is fairly constant over limited ranges.

Convection and Radiation of Heat by Irving Langmuir.

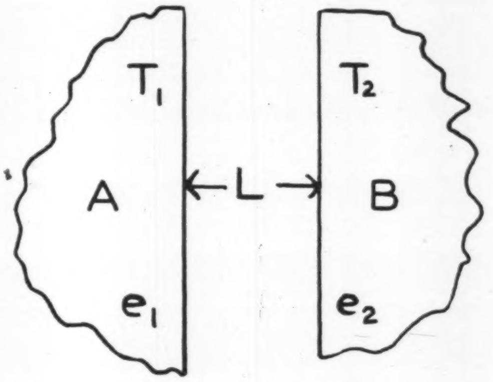

Fig. 1—Sections of Two Bodies Having Surfaces of Infinite Extent Parallel with Each Other

This paper was presented before the *American Electrochemical Society* in 1913 and contains formulae for calculating the heat transfer by radiation and convection. For radiation he uses the Stefan-Boltzman equation and for convection, he has developed the equation which for still air under average conditions may be expressed as

$$W_c = 0.296\ (T_1 - T_2)^{5/4}$$

where T_1 and T_2 represent the absolute temperatures of the surfaces enclosing the air space and W_c the rate of heat transfer by convection in Btu per square foot per hour.

Insulating Effect of Successive Air Spaces Bounded by Bright Metallic Surfaces, by L. W. Schad[2]. Two quite different test methods were employed for the purpose of obtaining data on the thermal conductivity of walls built up of air spaces separated by bright metallic screens. In addition some work was done to determine the convection and radiation constants of bright tin plate, the metal used for screens in this investigation. By substituting the numerical values of these constants in the usual heat transfer equations it was possible to check the results obtained by direct test and to determine the amounts of heat transferred by conduction through the air, by convection and by radiation.

The following is an abstract of this paper:

Let *A* and *B*, Fig. 1, represent sections of two bodies having surfaces of infinite extent parallel with each other and separated by a distance *L*. Assume that the space between the surfaces is filled with air and that the bodies are at temperatures T_1 and T_2 as shown. The rate of heat interchange is given by the following equation:

$$H = \frac{k_t\ (T_1 - T_2)}{L} + k_s\ (T_1 - T_2)^n + \frac{e_1 e_2}{e_1 + e_2 - e_1 e_2}\ (T_1^4 - T_2^4) \qquad (1)$$

where k_t is the thermal conductivity of air.
n and k_s are the convection constants.
e_1 is the radiation constant of body *A*.
e_2 is the radiation constant of body *B*.

For many insulating materials such as wood, corkboard, etc., the radiation constant e is high. Assuming a material with $e = 0.9$ of black body radiation, the overall conductivity coefficient was found to be 0.0097 watts per square inch for a temperature gradient of one degree Centigrade per inch which is equal to 2.65 Btu per hour per square foot for a gradient of 1 F per inch, and

6.6 per cent of the energy is transmitted by conduction,
28.9 per cent of the energy is transmitted by convection,
64.5 per cent of the energy is transmitted by radiation.

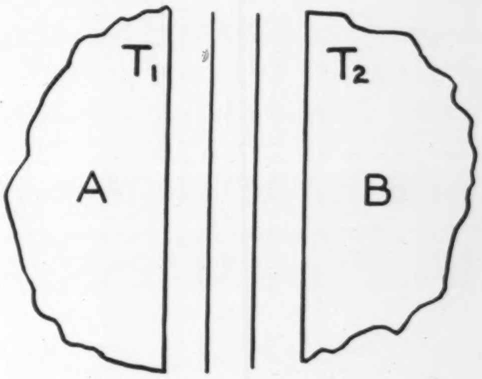

Fig. 2—Two Bright Tin Plate Screens Placed Between Bodies *A* and *B*

In this and the two following cases T_1 is taken as 373 deg Kelvin and T_2 as 273 deg Kelvin.

Substituting bright tin plate surfaces with $e = 0.1$ of black body radiation, the conductivity coefficient is found to be 0.0038 watts per square inch per degree Centigrade per inch and

15.8 per cent of the energy is transferred by conduction,
69.1 per cent of the energy is transferred by convection,
15.1 per cent of the energy is transferred by radiation.

As shown in Fig. 2 two bright tin plate screens have been placed between bodies *A* and *B*. The conductivity coefficient is now shown to be 0.00164 watts per square inch per degree Centigrate per inch and

43.4 per cent of the energy is transmitted by conduction,
43.4 per cent of the energy is transmitted by convection,
13.2 per cent of the energy is transmitted by radiation.

It is shown in the paper that n infinitely thin reflecting screens reduce the heat transfer by convection and radiation to approximately $\frac{1}{n+1}$ times the corresponding value without screens. Here it is assumed that the radiation constant e is the same for all surfaces in the heat path.

Two carefully planned experiments were carried out in the laboratory for the purpose of obtaining the conductivity coefficients of walls constructed along the foregoing principles. The usual method of guarded heater plate sandwiched between two similar samples under test with guard ring along the edges gave 0.00147 watts per square inch for a temperature gradient of 1 degree Centigrade per inch at an average temperature of 57 C. In this case three shields of bright tin plate were used in each sample and the air spaces were one-half inch thick.

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[2] Westinghouse Electric and Manufacturing Company.

TABLE 4—CONDUCTANCES OF AIR SPACES[a] AT VARIOUS MEAN TEMPERATURES

MEAN TEMP. DEG. F	CONDUCTANCES OF AIR SPACES FOR VARIOUS WIDTHS IN INCHES						
	0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

[a] *Thermal Resistance of Air Spaces*, By F. B. Rowley and A. B. Algren (JOURNAL, A. S. H. V. E., Vol. 35, No. 1, January, 1929).

A box having inside dimensions of 24 × 24 × 26 in. was first tested as a refrigerator and then as an oven. The walls of the box were 1.75 in. thick and three half-inch air spaces separated and enclosed by 1/16 in. bright tin plate. The following results were obtained:

Ave. temperature	*k_t in watts per square inch per degree Centigrade per inch*
17.5 C	0.00083
60.0	0.0013

Heat Insulation Facts, by L. B. McMillan. In a paper presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in 1920, Mr. McMillan makes the following statement:

"Next to perfect vacuum, the most effective insulation against the flow of heat is the minute confined air space. To be most effective, the air space must be absolutely enclosed and so small that circulation cannot take place within it, nor heat radiate across it to any appreciable extent. Even perfect vacuum would be ineffective for anything except very low temperatures unless it were broken up into small units or unless the surfaces were mirrored to prevent radiation as in the case of a thermos bottle. Therefore, the material which contains the greatest number of small confined air spaces per unit volume is the best insulator.

"Large air spaces are only effective when the difference in temperature across them is very small, because then convection is reduced to a minimum and the radiation at low temperatures is not great. However, at high temperatures, air spaces become very ineffective in preventing the flow of heat on account of more rapid circulation of air and the greatly increased radiation."

Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren. This investigation was conducted at the University of Minnesota in accordance with a cooperative research agreement between the University of Minnesota and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The results were reported in the January 1929 JOURNAL of the A. S. H. V. E. In discussing these tests, the authors make the following statement:

"There are several factors which affect air space coefficients. These are width of air space, mean temperature between surfaces, condition of surfaces and the ratio of the area to thickness of air space. The effects, due to width of space and mean temperature, should be the same for all surfaces. The surface effects will be different for different classes of materials, although in many cases it should be possible to group materials having similar characteristics. The effect of ratio of surface area to thickness will be more pronounced for small ratios, and it is probable that for average building constructions it may be neglected.

"In these experiments the air space coefficients were determined for different widths of air space and different mean temperatures. The surfaces used were Insulite, Masonite, Flaxlinum, Celotex, Compo-board and Gypsum board, the last two being paper covered.

"The results of this investigation show that the air space coefficients are the same for many materials, and undoubtedly for practical purposes, all materials used in building construction may be classified into a small number of groups, with characteristic constants for each group. With these constants and the proper external surface constants, it will be possible to calculate with accuracy the overall coefficients of heat transmission from the hot plate conductivities."

The conductances of air spaces at different mean temperatures, based on data reported in this article, are given in Table 4. These tests seem to be of more practical value than those of any other investigator. According to the results obtained, there is practically no increase in the conductance of an air space beyond about 1 in. in width. The average conductance of air spaces of this width or greater at a mean temperature of 40 F is about 1.10 Btu per hour per square foot per degree Fahrenheit difference in temperature, although this value is probably sufficiently accurate for all air spaces ½ in. or more in width. This conductance of 1.10 was used for computing the heat transmission coefficients of constructions containing air spaces in the A. S. H. V. E. GUIDE 1930.

The foregoing analyses comprise a few of the many investigations relative to the flow of heat across air spaces, which have been made in recent years. The results obtained in many of the other investigations are very interesting. Unfortunately, the scope of this article does not permit of an exhaustive analysis of these tests.

NOMINATIONS FOR OFFICERS OF THE SOCIETY FOR 1931

The Nominating Committee appointed to select candidates for Officers of the Society for the coming year, 1931, takes pleasure in submitting the following list of nominees:

For President:
W. H. Carrier, Newark, N. J.

For First Vice-President:
F. B. Rowley, Minneapolis, Minn.

For Second Vice-President:
W. T. Jones, Boston, Mass.

For Treasurer:
F. D. Mensing, Philadelphia, Pa.

For Members of the Council:

Three-Year Term

Roswell Farnham, Buffalo, N. Y.
E. Holt Gurney, Toronto, Ont.
E. K. Campbell, Kansas City, Mo.
E. O. Eastwood, Seattle, Wash.
D. S. Boyden, Boston, Mass.

Respectfully submitted,

NOMINATING COMMITTEE

G. L. Larson, *Chairman*

Chapters	*Representative*
Cleveland	H. M. Nobis
Illinois	J. F. Hale
Kansas City	N. W. Downes
Massachusetts	J. F. Tuttle
Michigan	J. H. Walker
Minnesota	G. C. Morgan
New York	H. B. Hedges
Western New York	O. K. Dyer
Ontario	A. S. Leitch
Pacific Northwest	E. O. Eastwood
Philadelphia	R. C. Bolsinger
Pittsburgh	T. M. Dugan
St. Louis	C. A. Pickett
Wisconsin	G. L. Larson

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. 9—Section 2. The Nominating Committee shall consist of a member designated by each Chapter, which shall effect its own organization and elect its own Chairman.

Section 3. The Committee may meet at the call of the Chairman, for discussion, but the official meeting shall be held at the Semi-Annual Meeting of the Society, at which time the candidates for each of the offices of the Society to be filled at the next Annual Meeting shall be selected by a majority vote of the members of the Committee present and voting. Five shall constitute a quorum.

Section 4. The Committee shall first secure the consent of all candidates selected, and shall, if possible, announce the names of the candidates at the Semi-Annual Meeting, but in any case the names of the candidates shall be certified to the Secretary of the Society at least four months before the next Annual Meeting. The Secretary shall publish these names in the October issue of The Journal.

Art. 10.—*Section* 3. The Secretary shall prepare ballots with the names of all candidates and forward them to the members at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election*. The election is governed by the By-Laws for the election of officers, with the single exception, that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee.

In accordance with the Regulations for the government of the Research Laboratory, adopted at the January, 1919, Meeting, and revised January, 1930, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January, 1931:

C. A. Booth, Buffalo, N. Y.
F. B. Howell, New York, N. Y.
Walter Klie, Cleveland, O.
F. B. Rowley, Minneapolis, Minn.
J. H. Walker, Detroit, Mich.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

Section I—Organization

Research Committee

1. There shall be a standing committee known as the Research Committtee, consisting of fifteen members, each serving for three years, and five retiring each year.

a. The Council shall nominate previous to October first of each year five members to fill the vacancies of those retiring at the next Annual Meeting.

b. The nominations made by the Council shall be published in the October issue of the Society's Journal.

c. Prior to December first of any year, any ten members, over their own signatures, may nominate one or more additional members of the Research Committee, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.

d. The election shall otherwise conform to the regulations provided for the election of officers of the Society.

e. Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

Willis H. Carrier

Willis Haviland Carrier, Newark, N. J., was born at Angola, N. Y., November 26, 1876, and his early education was acquired in the district schools of Erie, Pa., and Central High School, Buffalo, N. Y. In 1901, he was graduated with honors from Cornell University with a degree of M.E. Following his graduation he became a research engineer for the Buffalo Forge Co. and in 1906 attained the position of Chief Engineer.

In 1915, with a group of associates, The Carrier Engineering Corp. was founded, with Mr. Carrier as president, which position he now holds. Mr. Carrier, through his research and development work in the science of air conditioning and cooling, has received world-wide recognition for his discoveries and inventions which have contributed so much to the comfort and efficiency of the human race.

For the Society Mr. Carrier has contributed many notable papers and has served on many important committees. In addition, he has served on the Council for a number of years and has held the office of first and second vice-president. In other societies he has served with distinction as a member of the *A.S.M.E.*, as president of the *A.S.R.E.*, and has contributed to other professional and technical organizations.

Among his associates, Mr. Carrier is recognized as a scientist and is known as "The Chief." To the industrial world he is famed as a maker of weather and his host of friends know him as a royal good fellow, a keen fisherman and hunter and an enthusiastic golfer.

Mr. Carrier joined the Society in 1913 and has been nominated for president for 1931.

W. H. Carrier
Newark, N. J.

Frank B. Rowley

Frank B. Rowley, Minneapolis, Minn., was born April 3, 1882, at Evansville, Wis., and obtained his preparatory education in grade and high schools and his technical training at the University of Wisconsin. He was graduated with a degree of B.S. in 1905 and M.E. in 1906. After a year of engineering work with the Racine Gas Light Co. and the Power and Mining Machine Co. he joined the teaching staff of the Engineering Department of the University of Minnesota. During recent years, he has been Professor of Mechanical Engineering and Director of the Engineering Experiment Station, where many problems of heating and ventilating research have been investigated.

Professor Rowley's main interest has been in research and development work, particularly as it applies to heating and ventilating and he has acted as a consultant on many occasions. Professor Rowley in his college work was associated with Dean Allen, the first director of the A.S.H.V.E. Research Laboratory, and because of Prof. Rowley's outstanding work in research, it was fitting that he should serve on the Society's Committee on Research and as its chairman. He has handled other important assignments for the Society and is now second vice president as well as a member of the Council.

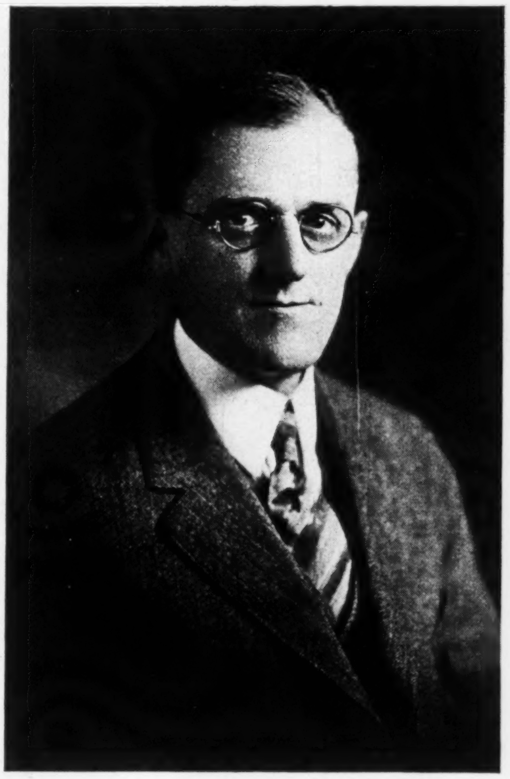
F. B. Rowley
Minneapolis, Minn.

Because of his familiarity with air handling problems and his study of engines, his interest in aviation naturally resulted and he has become an enthusiastic aviator and a licensed pilot. Recently he has been designated—the Complete Aerologist.

Professor Rowley became a member of the Society in 1918 and has been nominated for first vice-president for 1931.

William T. Jones

William T. Jones, Boston, Mass., was born August 14, 1883, at Peaks Island, Portland, Maine, and received his grade and high school education in Portland. In 1909 he was graduated from Lowell School and entered the drafting room of Whittier Machine Co. Following this he did design work for B. F. Sturtevant Co., and the Boston Elevated Railway Co.

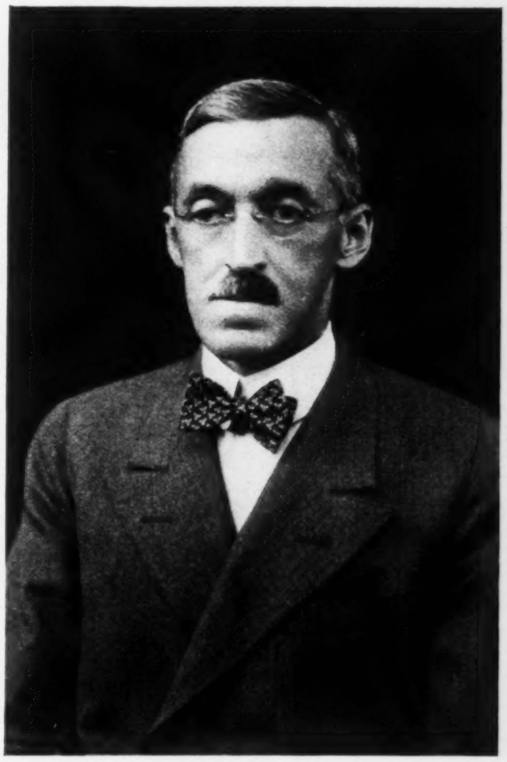


W. T. Jones
Boston, Mass.

In 1912 he started with Thermograde Valve Co. and this association lead to the Barnes & Jones partnership in 1921.

In the Society Mr. Jones has always been active in local chapter affairs and has served the Massachusetts Chapter as president and on its governing board. In 1925 Mr. Jones was elected to the Council, where he has served on several committees, and particularly on the Finance Committee, of which he was chairman in 1928 and 1929, and commencing this year was elected to membership on the Committee on Research.

Having been born at the seaside, Mr. Jones finds diversion as a sailboat skipper and other water sports. Mr. Jones became a member of the Society in 1915 and has been nominated as second vice-president for 1931.



F. D. Mensing
Philadelphia, Pa.

Frederick D. Mensing

Frederick D. Mensing, consulting engineer, Philadelphia, Pa., was born June 2, 1879, in Philadelphia, attended the public schools in that city and in addition was privately tutored. Mr. Mensing has specialized in steam and electrical engineering and after six years' service with the Philadelphia Bell Telephone Co., he did engineering work for Isaac H. Francis and Ballinger and Perit, before starting in business for himself as a consulting engineer. For many years Mr. Mensing has specialized as a consultant in the industrial field and has supervised the building and equipment of structures for manufacturing purposes. He has no particular hobbies or favorite sports, save smoking; for that, he always carries two pipes.

He has always taken an interest in local engineering activities, is a member of the Engineers Club of Philadelphia and the Philadelphia Chapter of the A.S.H.V.E., which he has served as president and on the board of governors. For the Society he has served as a member of many special committees, and in 1924 was chairman of the Guide Publication Committee. In recent years he has served as a member on the Committee on Research.

Mr. Mensing became a member of the Society in 1920 and has been nominated to serve as treasurer in 1931.

Local Chapter Reports

Kansas City

The Kansas City Chapter of the Society held its first meeting of the year 1929-30 in September, 1929. At this meeting papers on Gas, Its Supply, Distribution and Use were presented by Messrs. Koihn and Patten, engineers of the Kansas City Gas Co.

The members and guests attending the October meeting were honored by the presence of Thornton Lewis, President of the Society, A. V. Hutchinson, Secretary, and Cecil W. Farrar, chairman of the Committee on Increase of Membership.

The November meeting was devoted to Central Station Heating which was discussed in detail by David Caleb and John Arthur, of the Kansas City Power and Light Co.

The Effect of Air Space and Surface Coefficients on Heat Transmission was presented by Prof. F. B. Rowley, Minneapolis, Minn., at the December meeting.

An interesting report on the Engineering Conference in Tokyo and a general air conditioning discussion was given by W. H. Carrier, first vice president of the Society, which was thoroughly enjoyed by all present.

The February meeting was a general discussion on Unit Heaters; a report of the Annual Meeting of the Society, and also a discussion of the Boiler Code, was presented.

At the March meeting, A. W. Archer, architect, gave an interesting talk on the Mechanical Equipment of Bakeries.

The progress at the Society's Research Laboratory was very ably presented at the April meeting by F. C. Houghten, Director of the Research Laboratory at Pittsburgh. A rising vote of thanks was extended to Mr. Houghten in appreciation of his visit to the Chapter and for his interesting address which resulted in much discussion.

The May meeting was held at the Newbern Hotel, and was combined with the meeting of the Engineers' Club. Dean F. Paul Anderson delivered an interesting address to this unusually large gathering on the Aristocracy of Engineering, which was greatly appreciated by those present.

The June meeting of the Kansas City Chapter was a social affair consisting of golf, dinner and entertainment for both the members and ladies in attendance.

The officers for the Kansas City Chapter for the coming year are as follows:

President, E. K. Campbell
Vice-President, J. M. Arthur.
Secretary, W. A. Russell.
Treasurer, J. G. Lewis.
Board of Governors, Carl Clegg, F. P. Hitchcock and J. B. Fehlig.

At the September 1930 meeting the report of the Minneapolis meeting was given, together with special discussions on interesting papers which were presented at the summer meeting.

The October meeting has been set aside for a visit from L. A. Harding, president of the Society.

New York

The opening meeting of the season 1929-1930 for the New York Chapter of the Society was held October 21, 1929. Certified Heating in New York City was the topic of discussion which was given by W. E. Taylor, executive secretary of the *Heating and Piping Contractors National Association* in New York. M. E. Durkee, manager of the Certified Heating Bureau of the Association in New York, also spoke on that subject.

The next meeting of the Chapter was held on November 18, and was an exceptionally fine meeting. L. T. M. Ralston, consulting engineer, described the heating problems in connection with the construction of the Chrysler Building in New York City, which was most interesting to the members and guests present.

The December meeting was also a very interesting meeting, as the members of the New York Chapter were honored by having Thornton Lewis, president of the Society, among their guests. Mr. Lewis discussed the future of the A.S.H.V.E., which was thoroughly enjoyed. The Old and the New in Air Conditioning was described by Werner Nygren, consulting engineer.

At the January meeting, which was held on the 20th, W. J. Baldwin, Jr., chief engineer of the Commercial Department of the New York Steam Corp., spoke on the Steam Service from Central Stations in New York City, which was a very interesting topic.

The February meeting was of unusual interest, as W. H. Carrier, first vice president of the Society and past president of the New York Chapter, described in detail his recent trip to Japan, and illustrated his talk with motion pictures, which was greatly appreciated by the members and guests present. Heating Design as Taught and Practised was discussed in an address by F. W. Hanburger, special lecturer on heating and ventilating at the College of the City of New York. This talk was enjoyed by all those in attendance and a rising vote of thanks was extended to both Mr. Carrier and Mr. Hanburger at the conclusion of the meeting, which was well attended.

What the Heating Engineer has to Sell was the topic of discussion at the March meeting, held on the 17th, and attended by 65 members and guests. Dr. Harvey N. Davis, who delivered this interesting address, is the president of Stevens Institute at Hoboken, N. J. His talk covered the causes and effects of the results of the engineers' work in heating and ventilating the interior of buildings by the old methods, upon the heating and ventilating of New York City, outside of the buildings. He proved from physical laws that the exterior atmosphere is effected by present methods and would most assuredly have to be changed in the future. The gathering of data, such as dust count, smoke blanket and air pollution, was going forward at a rapid rate and would show that there existed a very real need for exterior ventilation.

On April 7 an inspection trip to the plant of the Nash Engineering Co., South Norwalk, Conn., was arranged. Following the inspection tour a shore dinner was served at the plant, after which the regular Chapter meeting was held.

The following officers were elected to serve the New York Chapter for 1930-31:

President, A. J. Offner.
Vice-President, Russell Donnelly.
Treasurer, F. E. W. Beebe.
Secretary, W. A. Swain.
Board of Governors, E. J. Ritchie, A. L. Baum and G. G. Schmidt.

The Meetings Committee was appointed and is composed of the following: A. S. Armagnac, S. L. Goodwin and W. J. Osborne.

The May meeting, one of the big meetings of the year, was held at the Building Trades Employers Club, New York, and was attended by 135 members and guests. The entertainment features were provided by Alfred Engle, who was responsible for the members and guests enjoying a delightful evening.

The 1930 season commences on October 20.

Philadelphia

The first meeting of the Philadelphia Chapter in 1930 was held in January and was devoted to the election of officers, and short talks by Thornton Lewis, at that time President of the Society, and H. P. Gant, past president. The officers elected at this meeting are as follows:

President, H. G. Black.
Vice-President, E. N. Sanbern.
Secretary, L. C. Davidson.
Treasurer, W. H. Wild.
Board of Governors, A. C. Edgar, M. F. Blankin and A. J. Nesbitt.

The February meeting was held at the Gratz Senior High School and J. D. Cassell was the speaker of the evening. Mr. Cassell selected as his topic the Mechanical Equipment Installed at the Gratz Senior High School, which was very interesting to all present.

R. V. Frost delivered an interesting discussion on the Proposed Revision to the Code for Rating Steam Heating Solid Fuel Hand Fired Boilers at the March meeting. Moving pictures were also presented at this meeting by Mr. Lang of the American Brass Co.

H. S. Christman of the Philadelphia Gas Works addressed the gathering at the April Chapter meeting on House Heating with Gas.

The May meeting was given over to the Annual Outing of the Philadelphia Chapter with golf and dinner at the Torresdale-Frankford Club. In the evening the members and guests were escorted through the J. J. Nesbitt Co. Inc. plant, and a short talk was given by Warren Ewald, research engineer of the company, on Noise Reduction in Ventilating Units.

During the months of June, July, August and September no meetings of the Chapter were held.

A tentative program has been arranged for the coming meetings for 1930, and it is expected that President L. A. Harding will address the November meeting. A talk on Air Conditioning in cigar factories and a visit to the Bayuk Cigar plant in Philadelphia has been planned for one meeting. It has been arranged to have W. H. Carrier, vice president of the Society, address the members and guests at another meeting.

The Philadelphia Chapter has a total of 199 members as of September 1, 1929, composed of 143 members, 42 Associate Members, and 14 Junior Members. In addition, there are five Limited Chapter members.

St. Louis

Many interesting meetings have been enjoyed by the members of the St. Louis Chapter of the Society during the past season, which began with the January 1930 meeting.

At this meeting Thomas Gale of the *Heating and Piping Contractors National Association* of St. Louis gave a very instructive discourse on The Heating and Piping Contractors National Standards, illustrating his talk with several examples for figuring radiation.

Quite an interesting discussion was entered into by the members present at the February meeting, at which F. J. McMorran, assisted by George Myers and C. W. Butler, discussed in detail the application of both floor type and suspended type unit heaters. At the conclusion it was agreed that the subject as presented by the speakers was most excellent.

E. A. Jones of the L. J. Mueller Furnace Co., Milwaukee, gave an instructive talk on Gas Heating at the March gathering. Considerable discussion followed, and numerous questions were asked.

The April meeting was turned over to E. B. Langenberg, who called on G. A. Helwig to read a paper on the Application of Warm Air Heating in Connection with the Industry. Mr. Langenberg also read a paper on the Application of Warm Air Heating in the Home. Both papers were well received by all those in attendance, and it was voted one of the most interesting meetings of the St. Louis Chapter.

At the May meeting the St. Louis Chapter was honored with the presence of Dean F. Paul Anderson of the University of Kentucky. Dean Anderson's subject was the Heating and Ventilating Engineer is Alive. He stressed the point that we are now passing through a stage of refinement of apparatus rather than one of new developments. His talk was most inspiring, and was thoroughly enjoyed by everyone present. At this meeting the Chapter was also honored by the presence of H. P. Gant of Philadelphia and past-president of the Society, and Dean A. Langsdorf of Washington University, St. Louis.

The June meeting was known as Ladies' Night, and an unusually large number of the ladies attended the meeting. Homer R. Linn, Chicago, spoke on Home Modernization, stressing the fact that the home to be modern must possess three important elements, namely: heating, plumbing, and lighting. At the conclusion of Mr. Linn's address, he was given a rising vote of thanks and the meeting was then turned over to the Entertainment Committee, who had arranged for bridge, in which all indulged. Many valuable prizes were awarded to the winners.

As has been customary, the St. Louis Chapter held no meeting during the months of July and August.

The September meeting, which was held on the 27th, was an outing at the North Shore Country Club. A golf tournament was arranged for the members and their guests, while the ladies enjoyed bridge, which was followed by dinner, served at the Club House, and dancing until the small hours of the morning.

Great progress has been made for the October Meeting and the St. Louis Chapter members are looking forward to a visit from the president of the Society, L. A. Harding, who is to speak on Utilization of the Sun's Energy.

At the November meeting it has been arranged that E. E. Carlson will deliver a discussion on Temperature Control, and C. G. Buder will talk on Oil Burners.

The December meeting will be in charge of T. J. Hester, who is preparing a paper on Heating Specialties. The St. Louis Chapter will also hold its annual election at this meeting, and will install the new officers for the year 1931.

The officers at present serving the St. Louis Chapter are as follows:

President, E. A. White.
First Vice-President, C. G. Buder.
Second Vice-President, R. M. Rosebrough.
Secretary to August 18, Paul Sodemann.
Secretary from August 18 to January 1, 1931, J. M. Foster.
Treasurer, A. L. Walters.
Board of Governors, F. J. McMorran, E. H. Quentin, L. Walter Moon and C. A. Pickett.

Southern California

August 12, 1930: The Southern California Chapter of the Society held its first regular meeting on August 12, at the University Club in Los Angeles.

Following the reports of the various committees, the Constitution and By-laws were discussed and adopted.

The officers elected to serve the Chapter for the coming year were then elected as follows:

President, O. W. Ott.
Vice-President, E. L. Ellingwood.
Secretary, H. B. Keeling.
Treasurer, Leo Hungerford.
Board of Governors, J. B. Armitage, J. A. Nelson and F. R. Winch.

The meeting then adjourned, and will convene on Tuesday, October 14, for the regular fall program of monthly meetings, which will be held on the first Tuesday after the 10th of each month.

Wisconsin

The members and guests of the Wisconsin Chapter of the Society assembled on September 30, 1929, for the first meeting of the season, 1929-30, at which the election of the following officers took place:

President, Fred G. Weimer.

Vice-President, John G. Shedron.

Secretary, V. A. Berghoefer.

Treasurer, Martin Erickson.

Board of Governors, Paul C. Downey, G. L. Larson and W. F. Noll.

At the October 21 meeting, the following committees were appointed by the president: *Membership Committee,* Howard Haupt, *Chairman,* A. Goethel and C. H. Jackson; *Picnic Committee,* E. A. Knab, *Chairman,* John Jung and A. Freeman; *Entertainment Committee,* Ernest Szekely, *Chairman,* E. A. Jones and C. H. Randolph. The speaker of the evening, A. V. Hutchinson, Secretary of the Society, was then called upon to speak on the various activities of the Society.

The third meeting of the season was held on November 18, 1929, and the members and guests present were very fortunate in having among their guests Thornton Lewis, Philadelphia, president of the Society, and C. W. Farrar, Buffalo, chairman of the membership drive. Mr. Lewis spoke on the future of the Society, while Mr. Farrar discussed the membership.

The following meeting was the December gathering, held on the 9th, and had as the speaker of the evening W. H. Schmidt, supervisor of Buildings of the State Department of Public Instruction. Mr. Schmidt's topic was Field Experiences and Observations of Public School Heating and Ventilating Systems.

E. L. Beckwith, president of the Detroit Stoker Co., delivered an interesting address at the January meeting on the subject of History, Development and Adaptability of the Stoker in the Heating Field, which was enjoyed by the members and guests in attendance.

The following meeting was held on February 17, 1930, with H. R. Linn as the guest speaker. Mr. Linn selected the Status of the Oil Burner in the Heating Industry for his subject, which was one of the most interesting and instructive talks of the season.

E. A. Jones, L. J. Mueller Furnace Co., and C. V. Haynes, Hoffman Specialty Co., were the speakers at the March 23 meeting. Mr. Jones discussed the Proposed Revision of the A.S.H.V.E. Boiler Code, while Mr. Haynes gave a general talk on what the Society has done and is doing.

The April meeting of the Wisconsin Chapter was held on the 21st of the month, and the members and guests enjoyed an interesting talk on the different Types of Heating and Ventilating Systems over a period of 50 years, which was rendered by J. F. Sprague of the Sprague Engineering Co.

The next meeting of the Chapter was held on May 17. At this meeting F. C. Houghten, Director of the Society's Research Laboratory at Pittsburgh, gave a very interesting resumé of the work at the Laboratory.

The final meeting of the season was in the form of a picnic and was held on June 9, 1930, and enjoyed by the members and guests who attended.

Haynes Selling Co. Moves

R. E. Jones, president of the Haynes Selling Co., Inc., announces the removal of their headquarters to 1518 Fairmont Ave., Philadelphia, Pa., where larger offices and show rooms will be maintained.

New Headquarters for Heating Association

According to the announcement of Allen W. Williams, managing director of the *National Warm Air Heating Association,* the general headquarters, Columbus, Ohio, have been removed to 3440 A. I. U. Building at Broad and Front Streets.

Schedule of Chapter Visits for President Harding

Beginning in October, President Harding will visit a number of chapters to speak on Utilization of the Sun's Energy, and he will be accompanied by C. W. Farrar, Treasurer, A. V. Hutchinson, Secretary, and F. C. Houghten, Director of the Research Laboratory. Mr. Houghten will speak on the Effect of Solar Radiation on Heating and Air Conditioning Problems, and C. W. Farrar will talk on Society Activities and Membership Aims.

The first stop will be in Chicago and on Monday, October 13, the Illinois Chapter will entertain President Harding and on the following night he will be the guest of the St. Louis Chapter.

On Wednesday, October 15, the Kansas City Chapter has made arrangements to greet Mr. Harding and he will then talk to the Wisconsin members on Thursday, October 16, and be with the Minnesota Chapter members on Friday, October 17.

Arrangements have been made for President Harding to visit the Philadelphia Chapter November 13, New York Chapter November 17, and Massachusetts Chapter December 1.

The plans of the various chapters indicate a large attendance of members and guests at these meetings.

Death of D. W. Chapman

On August 15, 1930, D. W. Chapman, of D. W. Chapman Co., consulting engineers, Baltimore, Md., died at the age of 57. Mr. Chapman was born on October 9, 1873, Canton, Ohio, and after graduating from the public schools, attended Ohio State University.

His entire business career was devoted to the mechanical engineering field, where he specialized in heating and ventilating. He was engineer and manager for the Chapman Steam Specialty Co., also president of the Chapman Plumbing & Supply Co., and his work covered the application of heating and ventilating apparatus and the plan, layout and specification for heating and ventilation and mechanical equipment of buildings, as well as structural and architectural work.

During the war Mr. Chapman served as a Captain of Engineers, from August, 1917, to October, 1918, in Washington, D. C., as an instructor and then as an ordnance gage designer. From October, 1918, to October, 1920, he was assigned to the Navy Department, first at Baltimore in charge of cargo ship repair and alteration, then on overseas transport service, and later in New York as chief inspector for U. S. vessels, then as steam engineering expert. His service rank was Ensign and Lieutenant, U.S.N.R.F. After serving for a time as inspector of steam engineering at Baltimore, Md., he had three months' service at sea as chief engineer of the U.S.S. Rapidan, before receiving his honorable discharge from the Navy Department October 1, 1920.

Upon leaving the government service he organized the D. W. Chapman Co., heating and ventilating engineers and contractors, Cleveland, Ohio. Later he moved his headquarters to Baltimore and also maintained offices at Washington, D. C., and in Tampa, Florida.

Mr. Chapman joined the Society in 1914 and the officers and Council regret the early passing of one of its loyal members and extend to his business associates and family a sincere expression of their sympathy.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 10 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by October 15, 1930, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

	REFERENCES	
CANDIDATES	*Proposers*	*Seconders*
Campbell, Walter Ernest, Sales Engr., American Radiator Co., Los Angeles, Calif.	O. W. Ott C. S. Anderson	A. B. Berg E. C. Evans
Johnson, Carl E., Sterling Electric Motors, Inc., Los Angeles, Calif.	H. B. Keeling C. S. Anderson	E. C. Evans A. H. Berg
Knox, James Ross, Asst. Heating Engr., Lord & Burnham Co., Irvington, N. Y.	V. F. Hooper A. P. Weiss	J. G. Kelly A. E. Bastedo
Oster, George Ray, American Air Filter Co., Louisville, Ky.	H. C. Murphy A. H. Berg	J. H. Milliken John Hellstrom
Peters, Herbert Henry, Branch Manager, L. J. Mueller Furnace Co., Salt Lake City, Utah.	E. A. Jones S. H. Berringer	P. E. Mueller W. F. Noll
Schoenoff, Alfred Emil, Student, Carnegie Institute of Technology, Pittsburgh, Pa.	S. E. Dibble	R. O. Granston C. O. Peterson
Ste. Marie, Gaston P., C. A. Dunham & Coy, Ltd., Montreal, Que., Canada.	G. L. Wiggs C. A. Thinn	A. J. Dickey M. W. Shears
Thornton, Frank, Jr., Manager, Residence Engrg., Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.	M. M. Timmerman E. Rasmussen	F. A. Gunther W. W. Stevenson
Tyson, Perry W., Htg. & Vtg. Draftsman, W. M. Anderson, Plumbers & Steamfitters, Philadelphia, Pa.	F. W. Wandless R. E. Jones	H. A. Erickson A. E. Kriebel
Winch, Franklin R., Consulting Engineer, 1031 S. Broadway, Los Angeles, Calif.	C. S. Anderson H. B. Keeling	A. H. Berg H. B. Keeling

Candidates Elected

In past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

Abboud, Alfred, Alfred Abboud & Co., Inc., Boston, Mass. (*Advancement.*)

Foy, Arthur T., Engrg. Dept., National Biscuit Co., New York, N. Y.

Hirst, James Noble, York Heating & Ventilating Corp., Philadelphia, Pa. (*Advancement.*)

Kozu, Tamiichro, Chief Engr., Daiwa Kogyo Co., Ltd., Kyobashi, Tokio, Japan.

Lees, William Douglas, Chief Electrician, Htg. & Vtg. Engr., Provincial Government of Alberta, Edmonton, Alta., Canada.

Scanlan, Chester J., Designing Engr., The Trane Co., La Crosse, Wis.

ASSOCIATES

Brown, W. Maynard, Asst. Secy., Warren Webster & Co., Camden, N. J.

JUNIORS

Anderson, Edwin L., Carrier Engrg. Corp., Newark, N. J.

Bailey, James Luther, Engr., Parks-Cramer Co., Charlotte, N. C.

Benson, John C., Carrier Engrg. Corp., Newark, N. J.

Coburn, Luther F., Carrier Engrg. Corp., Newark, N. J.

Darby, Marion Henry, Carrier Engrg. Corp., Newark, N. J.

Dowling, Joseph Michael, Carrier Engrg. Corp., Newark, N. J.

Edgar, James Struthers, Carrier Engrg. Corp., Newark, N. J.

Garneau, Leo, C. A. Dunham Co., Ltd., Toronto, Ont., Canada.

Gillham, John N., Carrier Engrg. Corp., Newark, N. J.

Haskett, Joseph Elwood, Carrier Engrg. Corp., Newark, N. J.

Mann, Lee Bernard, Carrier Engrg. Corp., Newark, N. J.

Meffert, George H., Carrier Engrg. Corp., Newark, N. J.

Paget, Francis King, Carrier Engrg. Corp., Newark, N. J.

Powers, Lowell Gable, Carrier Engrg. Corp., Newark, N. J.

Pulliam, William Gold, Carrier Engrg. Corp., Newark, N. J.

Skidmore, John Gardner, Carrier Engrg. Corp., Newark, N. J.

Smith, Frank J., Carrier Engrg. Corp., Newark, N. J.

Van Meter, Benjamin F., Carrier Engrg. Corp., Newark, N. J.

Walton, Louis A., Carrier Engrg. Corp., Newark, N. J.

Ware, James Elliot, Jr., Carrier Engrg. Corp., Newark, N. J.

Witmer, Charles Newcomer, Carrier Engrg. Corp., Newark, N. J.

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Heating·Piping and Air Conditioning EDITORIALS

Overall Cost of Maintenance

Maintenance! As Amos, of Amos and Andy, would say, "Now there's a word." It is looming up bigger and bigger on the industrial horizon from year to year. When the cost of producing goods has to be cut, who is the man to do it? Very often it is the maintenance engineer.

Leaks in maintenance costs are often obscure. There is a lowered efficiency of machines which may come so gradually as not to attract attention.

How long should a machine last? A premature replacement is just as much a loss as is continued operation of obsolete equipment.

What about the lowered morale of the operating force who must work with inferior equipment? That loss also is somewhat obscure; but tangible, nevertheless.

It was not always so. When slow speed, small capacity, cheap labor and cheap fuel were the order of the day, maintenance was not so much a problem. But with high speed, quantity production, expensive labor and costly machines, a shut-down is an important matter.

Different operating conditions and different economic conditions have compelled us to take an entirely new view of maintenance. Many industrial concerns now want to know what the overall cost will be for a period of ten years on all pieces of machinery or equipment, not merely the initial cost nor even the initial cost plus the cost of operation: they must know the initial cost, cost of operation and the cost of maintenance as well. The manufacturer, in years to come, who cannot give satisfactory information as to maintenance will be at a more serious disadvantage in competition with those who can supply this information than he is now.

A High Occupancy

This publication received a letter a few days ago calling attention to an office building which, in over forty years of operation, had vacancies which averaged no more than 10 per cent and in most cases much less than this amount.

Although office buildings are usually financed on the basis of a 10 per cent average vacancy, this building is said to show a remarkable record so far as office buildings go, and a statement of this kind at once arouses the curiosity of the reader as to what contributed to this low average.

Investigation reveals that in this instance there are many factors, of course, all of which come under "sound and progressive management." But, analyzing it further it appears that it is an outstanding example of good heating, air conditioning and modern mechanical services in general. A constant temperature is maintained during working hours. Constant studies have been made to promote economy in the power plant. Exhaust steam is used for heating except in early hours in cold weather. Good care is taken of all equipment. Each office is properly ventilated. All offices are inspected regularly. Tenants are informed of mechanical improvements.

There are, of course, the factors of elevators, light, location, attractiveness, etc. But if we are to learn anything from this building, it is that the careful work of the engineer has great influence upon the occupancy. This is an old building. It has less light than some newer buildings. Its value has not risen as fast as nearby properties have risen and it is believed that a leading reason for the occupancy of this building lies in the work of the engineer responsible for the mechanical equipment.

Whose Responsibility?

Meteorologists of the Mellon Institute place the annual soot fall in one large city as 1.031 tons per square mile. And that is not the worst city in the country in this respect. Economists have made what they believe to be conservative estimates of the cost of soot fall. The estimate for New York City for instance, is $20.00 per inhabitant per year. Similar figures are given for other cities.

Twenty dollars per person for New York City is a lot of money. There is an increasing agitation against soot and smoke; there are physicians who attack smoke and soot because of the physiological effect upon human beings, horticulturists because of the harm to plants and flowers, educators and eye specialists because of the increased eye strain, building operators because of the necessity for cleaning buildings outside as well as inside. There are many other groups who are taking an interest in smoke and soot abatement.

With this increasing agitation, responsibility is going to be placed on someone. It is recorded in the history of London that the chimney was blamed and a tax was put upon chimneys. We are prone these days to blame the owner of the chimney and the plant which makes the soot and smoke. It is significant, however, that the physicians, horticulturists, eye specialists, educators and building managers do not direct their complaints to the owner but to the manufacturer and the designing engineer. What then is the responsibility of the manufacturer and designing engineer in these premises?

Manufacturers and designers have done much to reduce smoke and soot; very often, the fault can be placed on the operator in charge of the equipment. Perhaps equipment will have to be developed that *can't* be handled incorrectly, if that is possible.

Placing The Heat

Dr. W. A. Evans, in his syndicated "How to Keep Well" column records the complaint of office employes in the matter of improper distribution of heat. He says, "In many offices during winter and spring, employes are complaining to the management that they are uncomfortably cold. The management, unmindful of the effects of discomfort of employes on the volume and the quality of work done, responds to the instinct to defend and points to the thermometer. This instrument set on an inside wall about seven feet above the floor, records eighty degrees. If the complaining employe is both well-informed and persistent he will appeal from the thermometer seven feet up to one at the floor level. But if the manager searches for a remedy he will have a hard time of it. He may strip his windows and close a few doors. He may change the location of some of his radiation so as to preheat some of the cold air before it can reach the floor. He may heat the floors or carpet or otherwise cover them. He may provide stools for the feet of those who sit or standing benches for those who stand as they work. The employes may wear warmer stockings or provide themselves with dry, warmer, looser, shoes to wear while working. And these are about all the remedies of which we know."

This is a condition which may be found in many offices. It is a major indictment of heating from the point of view of the heat-user. It is a most important problem to the industry. Because warm air will rise and cool air fall, it has been the general feeling that there is no real solution. As one engineer said "You cannot change the laws of nature." But while we do not change them, we utilize these laws and make them work for us.

Research now being conducted will eventually solve the problem completely; such research should receive the fullest co-operation.

Pipe Fittings

How dependent we are upon the honest workmanship that is put into common things; pipe fittings, for instance! Think of the billions of pipe fittings that are in use in hotels, apartments, schools, factories and industrial plants, streets, power houses and in numerous other places! That these enterprises continue to function is due to quite an extent to the skill and stability that is put into such fittings.

But dependability of pipe fittings was not acquired in a day. The invention of pipe goes back into dim and distant history. The pipe and its connections were among the first inventions of mankind. The invention of pipe and the necessary connections antedates that of the wheel. The first pipes were hollow reeds, such as bamboo, and joints were made by inserting the end of one pipe into another. Thus water was conducted for irrigation, for mixing of clay for bricks, and for other purposes. In succeeding centuries, pipe of terra cotta and of metals was used but from that time to the present the connection has been a vital part of the pipe line.

To record the various inventions for pipe connections would require a volume. Recently fifty-five different ones were illustrated in one magazine article. Such a record should reveal to us the fact that the pipe connections of the present day are one of the most serviceable inventions of mankind.

For these reasons and because we expect stability in pipe connections, care should be taken in their fabrication.

A manufacturer, writing to this publication recently, says "The real difficulty contractors seem to be having is in fittings being off-standard. We have encountered plenty of real grief because of off-standard fittings, and the necessity of having them conform to the standards should be emphasized to every one."

Even were jobs designed for high pressures and temperatures not going in every day, the manufacturer's point would be one to stress emphatically. Off-standard fittings mean waste to all concerned —to the piping contractor in time lost on erection and to the manufacturer in lack of confidence in his product.

To let pipe fittings supply their important function in industry today, they should be made to conform rigidly to the standards. In this way, they will fulfill their need and insure a growing usage.

INTERVIEWS OF INTEREST

New Plants for Old at Armour's Pay Costs—and More—in Two Years' Time

Certain changes and additions made in one of Armour & Company's power plants are of particular interest to engineers: (1) Because they seem to show advantage in a consistent policy to concentrate apparatus and to reduce within the plant the numbers of distribution points for power; and (2) because accumulating economic proof seems to indicate so definitely the quick and sure returns from modernization programs.

"Maintenance with us," states O. A. Anderson, chief of Armour's engineering division, "is subjected to budgetary control. In every plant each year there has been a continuous increase of consumed power and multiplied demand upon hot water supply services for cleansing purposes and for process work.

"Costs mount in marked degree as plants grow old and it becomes a problem to determine how long a plant shall live, and when it should be retired from service.

"Rehabilitation, too, is budgeted in orderly fashion in all our work. We always have the advantage of full perspective in repairs we do. Plants new and old are in operation side by side. We contrast costs and operating conditions in similar plants and processes from week to week. The gains we make by change in power control or distribution measure are soon reflected in our records, and we can tell very definitely where we should substitute new plants for old and new plans for old.

"The things we did at this one plant are typical. Remodeling work included:

(1) Purchasing our power from an outside source;

(2) Installation of electrically-operated equipment;

(3) Installation of an electric ice machine—the second unit of a plan that contemplates the use of electric refrigeration. The high speed compressor unit now in place at Omaha occupies not more than one-fourth of the space required for the same capacity in older plants. Service and economy is excellent and investment is low. Sizes of plant exhaust steam lines are greatly reduced.

(4) A further change that conforms to practice elsewhere was the installation of a modern type of shell and tube ammonia condensing equipment.

(5) The master hot water supply system here, another typical Armour development, is heated with exhaust steam from the remaining steam units in the plant. It reflects our present policy to employ large master water heating systems instead of multiple, smaller units.

"The net cost of changes made in the power plant approximated $120,000. The installation work was completed in August, 1929. The net result of the comprehensive planning has been that we could shut down one extra boiler room that formerly had to be maintained on the line. We save in organization difficulties, and use of steam is more economical.

"The evidence is plain. Thirty-nine weeks of continuous operation have shown an actual saving over earlier schedules of about $70,000. Notwithstanding the fact that the ensuing months showed a larger 'kill,' and heavier process load, the net saving by the end of 1930 will be at least $100,000, and this estimate is made before any figures are compiled to reflect the saving that results from installing a master system of hot water supply. The re-vamping done will thus be able to liquidate the expense because of savings made in less than two years.

"Our concentration of power plants will continue. We purchase excess power instead of generating it wherever satisfactory local arrangements can be made. We look to better functioning of the plant, and, in general, tend toward reduction in remote control.

"There is not much change in piping, but increasingly we do welded work, with quite a sprinkling of fabricated jobs delivered complete upon our specifications."

Power plant economy is a problem of heat balance. The effort, then, is to generate no steam beyond plant uses. Low pressure steam for heat, hot water, and for processes within the plant accounts for all exhaust. Power costs from outside sources are cheaper than production unless waste heat is extracted for useful work within the plant.

This means that cost factors in plant maintenance must compete with purchased power from public utilities. It makes for careful operation and supervision. New plants are interesting to plan on such a basis, but predicting economic results becomes more difficult when one must consider these factors in an ageing plant and know just when it is justifiable to retire equipment still capable of further use.

Each expenditure for routine work or a new piece of machinery becomes a major engineering project on this basis. In the packing industry about 35 per cent of the total steam generated in the boiler room has sufficed for power service in the plant. The remaining 65 per cent must find economical application in plant and process work.

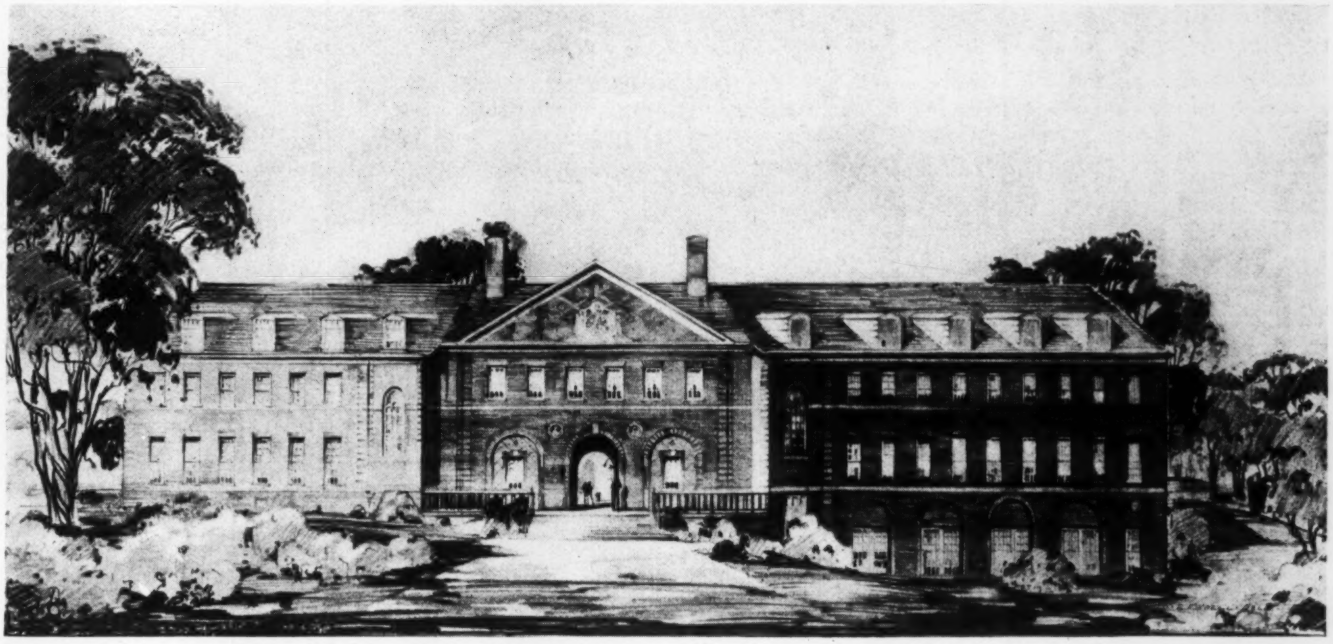
New uses for low pressure steam have been evolved. Live steam is jacketed or piped and kept in closed systems rather than used in the old manner of direct application to the products to be cooked. Water is heated up to 190 deg. and stored in a central tank for circulation rather than relying upon live steam to heat cold water in isolated tanks. Control devices keep down excess heat, and plant management obviates peak loads that in the past called for extra boilers or uneconomical loads.

All this reflects in engineering design, and tuning up an old plant may represent a bigger problem than the design of a new one.

New Building for Beloit

In Morse-Ingersoll Hall at Beloit College, Beloit, Wisconsin, school room radiation is calculated only for heat leakage. Incoming ventilation air can fluctuate through a very wide temperature range (from 70 to 90 F) and any drop in temperature at the radiators, instead of turning on more radiator steam (through the temperature control system) results in warmer ventilation air. The incoming air, instead of steam, varies to maintain the necessary inside temperature.

The auditorium is interesting in that the wall inlets for tempered air are supplemented by a horizontal mushroom arrangement beneath the front row of seats, which deflects the incoming air forward over the platform to heat the stage and forward portions of the room.



Morse-Ingersoll Building, Beloit College, Granger & Bollenbacher, Architects

Granger and Bollenbacher, Chicago, are the architects of this building, and Ernest V. Lippe, engineer with that organization, is the engineer on design. The building has been erected at a total cost of 35 cents per cubic foot.

Three separate units, for zone heating, are included in the ventilation plan. The manner in which this flexibility has been achieved in combination with reduced costs is explained by Mr. Lippe:

"The air supply is on the standard basis of 30 cubic feet of air per person per minute in occupied rooms," he states. "Our arrangement of three systems of air ducts permits sectional use of the building. Economical power reduction can be effected in little-used portions of the building, or one part can be used independently of others.

"Exhaust is mechanical. It could be gravity. Office rooms have no mechanical ventilation. Humidity control is automatic and separately operated steam nozzle humidifiers supply the three duct systems.

"The plan used here requires much less radiation than is the rule. Recirculation is provided for when rooms are not occupied, at night and during early morning hours. Exhaust air is carried up to the attic and collected there to be exhausted by three fans, one serving each system.

"Two important considerations must be met in school room ventilation: (1) Positive air change must be effected, with comfort, and on a calculated basis; and (2) cost relationships must in every case be kept within rational and consistent figures.

"Flexibility in this building has not been sacrificed to lower cost considerations. All excess radiation has been carefully eliminated. No sacrifice of air change could be made. Zoning the building and supplying a somewhat wider range than usual in the temperatures of the incoming air has enabled us to maintain a sensitive response to outside fluctuations with economy.

"Morse-Ingersoll Hall will house facilities for academic work in languages and science. The class rooms have varied uses. The work plan and mechanics have been well adapted here. The saving is considerable. Zone ventilation is quite as practicable as zone heating. Unit humidification, too, is accomplished."—*S. P. M.*

Marks' Handbook Revised

The third edition of the "Mechanical Engineers' Handbook" includes much additional material because of the trends toward higher pressures and temperatures, increasing efficiencies and capacities, etc. Technical discussions of such trends serve to supplement the large amount of physical and test results published. While the book contains over two hundred more pages than did the 1924 edition, its bulk is not increased, as a thinner paper stock has been used.

New steam tables for the high pressure region are included in the added sections of this standard reference work.

The book, which is published by the McGraw-Hill Book Company, 370 Seventh ave., New York City, is edited by Lionel S. Marks. It has over 2,000 pages, more than a thousand illustrations and diagrams, and is bound in a flexible cover. Its size is 4½ by 7.

PRACTICAL PIPING PROBLEMS

Keeping Hydraulic Packing Ready for Use
Oil in Oxygen Piping Dangerous

by
W. H. WILSON

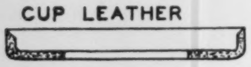
Care of Packing

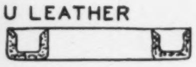
In the hydraulic piping field, in connection with the installation of and maintenance of the pipe work as used and applied to hydraulic presses, riveters and other hydraulic equipment, the packing is one of the important parts of the maintenance engineer's work. Continuous and uninterrupted operation requires that the packing be carefully applied.

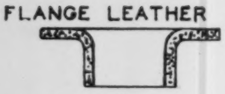
Hydraulic leathers, cup, U, and flange shapes, are used extensively for packing and operating valves, especially the spindles of the poppet type valve. Leathers are also used for the prefill, filling check valves, some types of stop valves (as for an example, the balanced type) pilot valves and various types of valves used in connection with the piping required for the operation and control of hydraulic machinery. Pipefitters engaged in this work should give careful consideration to the hydraulic leathers.

These types of leather packing should be of material of good quality. This, for that matter, should be true of all packing used for hydraulic cylinders, rams, plungers and valve stems, as the cost of the packing is a small item when compared to the labor required for replacing it, which in many cases is a large and, frequently, an overtime job.

The amount of working time lost while the hydraulic press or other equipment is shut down for this repair work is a large item, especially when the workmen operating the equipment are waiting for the packing to be replaced before they can resume their work, together with the delay in the working schedule and consequent loss of production. The first cost of the packing is generally the smallest item and one small leather can cause a delay and is expensive to replace.







Several Forms of Hydraulic Leathers

The first thing to consider is material of a good quality. Manufacturers of the better grades of leathers for this purpose usually use imported hides from France, Spain, Switzerland and other countries, where the cattle, it is said, receive good care and are not sent to market until they are older and their hides are thicker and tougher than that of the younger beef cattle.

The thickness of the leathers used for hydraulic packing is usually about 3/16 of an inch.

The next consideration is leathers that are well molded and accurately made to the proper shapes. Maintenance men frequently have molds made and prepare their own leathers. Manufacturers of hydraulic leather packings supply leathers accurately molded and properly treated; these leathers are often preferred by many men engaged in hydraulic repair work.

In connection with the use of leathers, it is evident that the hydraulic rams, cylinders, plungers and valve stems should be in good condition, accurately machined and properly assembled, as leathers could not be expected to hold and give service for any reasonable period of time, if the parts of the equipment that come in contact with the leathers are badly scored, improperly fitted or if they pinch the leathers.

Heat is also detrimental to leathers. This should be kept in mind when hydraulic presses are used for pressing out shapes from hot steel. Stripper or kick out rams, packed with leather, sometimes require frequent renewal of leather packing in such cases. To overcome this trouble, the design of the cylinder and ram should be changed, if possible, to keep the leather away from the heated material.

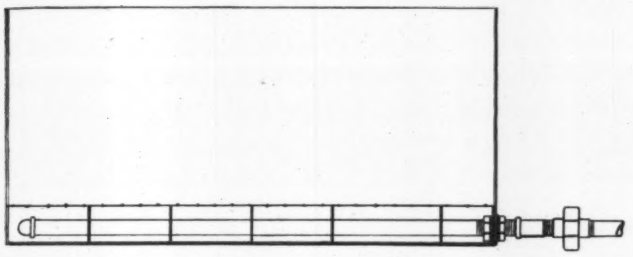
An important item in connection with the use of hydraulic leathers is their care before use. The manufacturers of these leathers usually furnish them treated or coated with a preparation that renders them ready for use.

Efficient operation of hydraulic machinery makes it necessary to keep leathers of the various sizes used, on hand, in order that repairs can be made quickly and delays avoided. It usually requires quite a number of different sizes of leathers, small ones and large ones, different shapes and forms, to be kept on hand for use in an emergency. Some sizes may be kept on hand for a year or two before they are used but when needed they are wanted quickly. When the leathers are stored in tool boxes, store rooms or bins, for any length of time, they become hard. When put in operation it takes some time for them to get adapted to the surfaces with which they come in contact and they will not last as long as fresh leathers. Leathers of this type sometimes crack and break shortly after being applied.

Keeping Leathers in Good Condition

Treating a leather with tallow and beeswax will not only prolong its life but will make it much easier to install. In connection with this, attention is called to the following method that has been used to advantage by some users of hydraulic leather packings:

Procure a steel box of a suitable size that will hold all of the leathers that are required to be kept on hand. For the ordinary sizes of leathers used, a steel box 18 inches long and 24 inches wide, of a suitable height, is ample. Put in this box a perforated or coarse wire mesh support, about 2 inches from the bottom. If steam for



Steel Container for Keeping Leathers in a Wax Compound Until Ready for Use. The Side View Is Shown Here

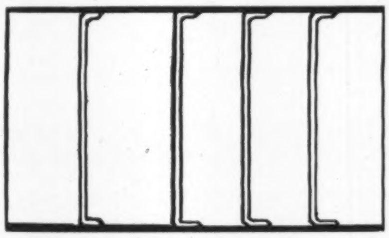
heating is available a pipe coil can be placed under this wire mesh support. If furnaces are used in connection with the hydraulic equipment, dispense with the pipe coil and set the box on or near a furnace so that the heat will reach it.

Put into this box forty pounds of mutton tallow, ten pounds of beeswax and a quart of neat's-foot oil. Then melt the mixture and place the leathers in it. Remove the box from the furnace, or if a steam coil is used for heating, shut off the steam; the mixture will harden and the leathers will be preserved and remain in good condition until ready for use.

When occasion requires a leather to be replaced, turn on the heat and proceed to remove the valve spindle or do what is necessary to prepare for the replacement and by that time the mixture should be melted. Pull out of the box the proper leather, which will be well lubricated, and put it in place. Then allow the mixture to harden and the remaining leathers will remain in storage until another one is needed.

Forms for Storing Leathers

Keeping the leathers in a form, before being put in service, is good practice, as it prevents them from getting out of shape. Forms for storing these leathers can sometimes be made out of a piece of ordinary pipe. For a cup leather, 8 inches outside diameter, use a short piece of 8-inch steel or wrought iron pipe. If not exactly the right inside diameter, a little lathe work will bring it to the proper dimensions. Place the leathers inside of this piece of pipe, using it as a container. This container, with the leathers, is then placed in the box containing the mixture of tallow and beeswax and the leathers will be stored, treated and kept in good condition, until ready for use. The proper size then can be easily found, when needed.



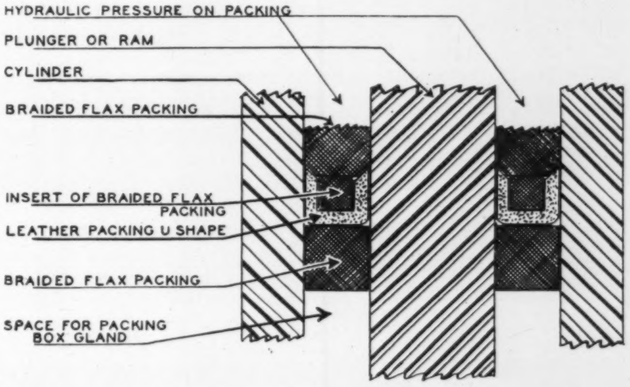
Leathers Stored in a Cylindrical Container

Use of Braided Flax

Leather packing is sometimes used in connection with braided flax for packing large hydraulic rams and plungers. The U type of leather, with a piece of braided flax, inserted in the opening of the U leather, is used in some cases. Opinions differ as to the arrangement and the use of different kinds of packing for hydraulic equipment.

The following method gives good results, for large hydraulic rams, using water at atmospheric temperatures and 5,000 pounds working pressure.

The first ring of packing used is of fine copper wire, braided, the ends of this packing seized with fine copper wire and then dipped in hot solder. This kind of packing has considerable "stretch" and the measurement should be taken carefully to get the length required using a butt joint. The next ring should be a good grade of duck packing. Duck packings for high pressures are usually hard and require soaking in warm water to render them pliable before putting them in place. From the way this packing is usually made, it can and should be applied with a step joint. Next use a ring of a good grade, long fiber, hard braided flax, using a miter joint. Next a U leather packing with an insert of braided flax, as previously mentioned, using a butt joint.


HYDRAULIC PRESSURE ON PACKING
PLUNGER OR RAM
CYLINDER
BRAIDED FLAX PACKING
INSERT OF BRAIDED FLAX PACKING
LEATHER PACKING U SHAPE
BRAIDED FLAX PACKING
SPACE FOR PACKING BOX GLAND

Hydraulic U Leather Used in Connection with Flax Packing. Detail of Position of Packing Before Packing Box Gland Is Drawn Up Tight

The packing box on large cylinders should allow room for from six to eight rings of packing to be used.

Hard braided flax packing is often used below the U leather ring.

Before inserting the flax packing in the U leather packing, a small amount of the tallow and beeswax mixture put inside of the U leather will help to lubricate the packing after it is in use.

Clean Oxygen Piping

Modern industrial practice requires, in some cases, pipe lines for supplying oxygen, which is used in shop work, usually in connection with acetylene, hydrogen and other gases. Oxygen is expensive; therefore, it is essential that the pipe lines be free from leakage. The use of a good grade of pipe, with well made joints, is essential.

Where it is necessary to connect to valves by using threaded joints, a mixture of litharge and chemically pure glycerine applied to the threads will aid materially in making a tight joint. Valve stems, such as line stop valves, should be packed with a good grade of fiber packing which has been treated with pure glycerine. This packing will dry out in time and should be renewed periodically, especially when the valves are frequently used. The manufacturers of oxygen equipment supply valves